

Gas absorption heat pumps in the built environment

Internship project at Royal Haskoning

Energetic feasibility study on using gas-fired absorption heat pumps in buildings needing much more heating than cooling. Models of a gas absorption heat pump and a building heating system are built and tested.

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Abstract

Electrically driven heat pumps are an established means to provide heating to office buildings. When these are coupled to geothermal energy storage, they are also able to provide cooling to the building. This concept is unfortunately not suited to buildings that need much more heating than cooling, like nursing homes.

Gas-fired heat pumps are a relatively young development in the built environment. Since these heat pumps have higher heat-to-cold ratio than electrically driven heat pumps, they seem a much better solution to meet the requirements of buildings like nursing homes.

The goal of the present project is to find out whether gas-fired heat pumps are a good means to lower the energy use for heating and cooling of buildings with a high ratio of heating demand to cooling demand. The project focussed on gas absorption heat pumps and nursing homes.

A detailed thermodynamic model of a typical gas absorption heat pump (GAHP) for the built environment was developed in Matlab/Simulink. This was done in order to gain more understanding on gas absorption heat pumps and to find out whether the efficiency values given by their manufacturer are reasonable. The results from this modelling have shown that the efficiency values given by its manufacturer are physically attainable.

In buildings, heating systems commonly consist of heat pumps, gas-fired boilers and a buffer vessel. A model of such a heating system using GAHP's was built. Using the derived heat demand profile for a typical 3000 m² Dutch nursing home, simulations for one year were performed. The results from the simulations have shown that for heating the best energetic performance was obtained using a heating system employing a water source gas absorption heat pump. This type of heat pump is coupled to geothermal energy storage in underground aquifers.

To consider both heating and cooling, five different heating and cooling concepts were envisioned. All concepts make use of gas-fired boilers for backup heating. The first three concepts use air source, ground source and water source GAHP's, while the fourth uses a water source vapour compression heat pump. Furthermore, the concepts using GAHP's also use an air-cooled chiller for supplementary cooling. The fifth system is a gas-fired boiler combined with an air-cooled chiller. The results from the calculations have shown that the lowest primary energy use was obtained for the water source GAHP concept, followed by the ground source GAHP concept. The gas-fired boiler system showed the worst performance.

It can thus be concluded that of the concepts considered for nursing homes a water source gas absorption heat pump, combined with gas-fired boilers and an air cooled chiller provides the best means to lower the buildings primary energy use for heating and cooling.

Samenvatting

Door elektriciteit aangedreven warmtepompen zijn een gevestigde technologie voor het verwarmen en koelen van kantoorgebouwen. Wanneer deze warmtepompen zijn gekoppeld aan warmte- en koude opslag in de bodem (WKO), dan zijn ze ook in staat om de koeling voor het gebouw te leveren. Dit concept is helaas niet geschikt voor gebouwen die veel meer verwarming dan koeling nodig hebben, zoals verzorgingstehuizen.

Door gasvormige brandstoffen aangedreven warmtepompen zijn een relatief jonge ontwikkeling in de gebouwde omgeving. Aangezien deze warmtepompen een hogere warmte-koude verhouding hebben dan elektrisch aangedreven warmtepompen, lijken ze een veel betere oplossing te zijn voor het voorzien in de behoefte van gebouwen als verzorgingstehuizen.

Het doel van dit project is om erachter te komen of door gasvormige brandstoffen aangedreven warmtepompen een geschikte technologie vormen om het energieverbruik van gebouwen te verminderen die veel meer verwarming dan koeling nodig hebben. Het project richt zich op gas absorptie warmtepompen en verzorgingstehuizen.

Er is een gedetailleerd thermodynamisch model gemaakt in Matlab/Simulink van een typische gasabsorptie warmtepomp (GAWP) voor de gebouwde omgeving. Het doel hiervan was om het begrip van de gasabsorptie warmtepompen te vergroten en om erachter te komen of de door de fabrikant opgegeven waarden voor de efficiëntie van deze systemen fysisch haalbaar zijn.

Verwarmingssystemen voor gebouwen bestaan veelal uit warmtepompen, gasketels en een buffervat. Er is een model van een dergelijk verwarmingssysteem gemaakt, met daarin GAWP's. Simulaties zijn uitgevoerd voor een jaar, waarbij een afgeleid warmtevraag profiel voor een typisch Nederlands verzorgingstehuis met een oppervlak van 3000 m² is gebruikt. De resultaten van deze simulaties hebben aangetoond dat de beste energetische prestaties voor verwarming worden bereikt met een verwarmingssysteem dat gebruik maakt van een GAWP met WKO.

Om zowel verwarming als koeling te beschouwen zijn er vijf verschillende concepten voor verwarming en koeling bedacht. Alle concepten maken gebruik van gasketels voor ondersteuning bij piekbelasting. De eerste drie concepten gebruiken GAWP's met lucht, grond of WKO als warmtebron. Concept vier gebruikt een damp compressie warmtepomp. Verder maken alle concepten met GAWP's gebruik van een luchtgekoelde koelmachine voor extra koeling. Het vijfde concept is een gasketel met een luchtgekoelde koelmachine. De resultaten van de berekeningen hebben laten zien dat het laagste primaire energiegebruik wordt bereikt met het GAWP concept in combinatie met WKO. Dit systeem wordt gevolgd door het GAWP concept met grond als bron. Het concept met de gasgestookte ketel laat de slechtste energetische prestaties zien.

Al met al kan worden geconcludeerd dat van de voor verzorgingstehuizen beschouwde concepten het concept met de GAWP met WKO, gecombineerd met gasketels en een extra koelmachine de beste manier vormt om het primaire energiegebruik voor verwarming en koeling te verminderen.

Introduction

New office buildings not using heat pumps are becoming an exception these days. Engineers designing the climate installations for buildings need to have a substantial amount of knowledge in this field in order to select the right installation for a certain application. Since heat pumps are a lot more complicated than gas-fired boilers and a lot of different types are available on the market, this is not an easy job. The advent of gas-fired heat pumps makes matters even more complicated. In order to keep up with these new developments Gerard Jansen from the Building Services department wrote an internship assignment proposal that has resulted in the present report.

The reason that Gerard is interested in gas-fired heat pumps is that the amount of heat they deliver is much higher than the amount of heat they extract at their evaporator. The fact that this heat-to-cold ratio is much higher than for electricity driven vapour-compression heat pumps makes them particularly interesting for buildings that need much more heating than cooling. A good example of such a building is a nursing home. In order to be able to make use of this advantage, the heat pump has to be coupled to heat and cold storage in underground aquifers or to a vertical ground loop heat exchanger.

The goal of the present project is to find out whether gas-fired heat pumps are a good means to lower the energy use for heating and cooling of buildings with a high ratio of heating demand to cooling demand. In order to keep the assignment within the limits of what can be achieved in a three-month internship period the decision was made to focus exclusively on gas absorption heat pumps for the supply side and nursing homes for the demand side. More details on the goal and scope of the project are given in Chapter 1.

In order to get a clear overview of the current situation of heat pumps in the built environment a general literature study on this topic was carried out first. The results from this study can be read in Chapter 2, together with some basic explanation on the thermodynamic principles underlying the different heat pump cycles.

Because no study project can be successful unless the studied topic is thoroughly understood, a detailed thermodynamic model of a gas absorption heat pump was built and tested against manufacturer data. The modelling approach is explained in Chapter 3, the results are shown and validated in Chapter 4.

In reality a heat pump is never on its own. Rather it is part of a complete building heating system, commonly employing gas boilers for covering load peaks. In order to take this situation into account a heating system model incorporating gas absorption heat pumps was built. The modelling process is described in Chapter 5 and the models are validated in Chapter 6. The results from the models for an air source heat pump, a ground source heat pump and a water source heat pump are given in Chapter 7.



As described above, the goal of the present project is to find out whether gas-fired heat pumps are a good means to lower the energy use for heating and cooling of buildings with a high heat-to-cold ratio. Finding the answer to this question using the previous modelling results is the topic of Chapter 8.

The next chapter is the discussion. Here the results are summarized and accounted for. Furthermore the limitations of the models used are discussed. After the discussion, all the relevant conclusions from the report are bundled in the conclusion. After the conclusion recommendations for future work are given in the last chapter of this report.



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Nomenclature

Roman

$c_{p,i}$	specific heat capacity of component i [kJ/kg·K]
E	electrical energy [kWh _e]
$\dot{E}$	electrical power [kW _e]
E_{ratio_pumps}	ratio of pump electrical energy to heat delivered to the building heat exchanger or heat pump condenser [kW _e /kW _{th}]
f_{def}	defrost frequency factor [hr ⁻¹]
f_{nd}	night-day heating power factor [-]
h	specific enthalpy [kJ/kg]
m	mass [kg]
$\dot{m}$	mass flow [kg/s]
P	pressure [bar]
q	vapour quality [kg vapour/kg vapour-liquid]
Q	heat/cold [kWh _{th}]
$\dot{Q}$	heating/cooling flow [kW _{th}]
Δt	time step [various units]
t_{def}	defrosting duration [hr]
T	temperature [°C, K]
UA	heat transfer coefficient times heat transfer area [kW _{th} /K]
v	specific volume [m ³ /kg]
$\dot{v}$	volume flow rate [m ³ /s, m ³ /hr]
V_s	buffer vessel hot water storage volume [m ³]
$\dot{W}$	mechanical power [kW _e]
$\dot{W}_c$	compression power [kW _e]
x	liquid phase mass fraction [kg ammonia/kg solution]
y	vapour phase mass fraction [kg ammonia/kg vapour]

Greek

η	efficiency [-]
$\eta_{el,grid}$	electrical efficiency relating the electrical power drawn from the grid to the power plant thermal fuel input [kW _e /kW _{th}]
ρ	density [kg/m ³]
$\varphi_{v,heating}$	volumetric heating capacity [kJ/m ³]
Δ	differential [various units]



ε	heat exchanger effectiveness [-]
<i>Subscripts</i>	
<i>abs</i>	absorber
<i>abs,out</i>	leaving the absorber
<i>air</i>	outdoor air
<i>aux</i>	auxiliaries
<i>balance</i>	balance between heat supply and demand
<i>boiler</i>	natural gas-fired boiler
<i>burner</i>	referring to natural gas or LPG burner
<i>building,in</i>	entering the building
<i>building,out</i>	leaving the building
<i>B</i>	building
<i>comp</i>	compressor
<i>cond / c</i>	heat pump condenser
<i>cond,in</i>	entering the condenser
<i>corrected</i>	corrected for overprediction by the manufacturer
<i>crit</i>	at the critical point
<i>d</i>	day
<i>evap / e</i>	heat pump evaporator
<i>el</i>	electrical
<i>ERH</i>	electrical resistance heating
<i>fuel</i>	unspecified type of fuel
<i>fuel, power _ plant</i>	fuel, burned at a power plant
<i>gas</i>	natural gas fuel ending up as <u>useful heat</u> absorbed by to the system
<i>gas, fed</i>	natural gas <u>fed</u> to the system
<i>gen</i>	generator
<i>gen,vap,out</i>	vapour leaving the generator
<i>gen,weak,out</i>	weak solution leaving the generator
<i>GE</i>	gas engine
<i>GE _ used</i>	heat from the gas engine used for heating the building
<i>heating</i>	heating during normal operation
<i>heating,defrost</i>	heating during defrosting operation
<i>HP</i>	heat pump (excluding auxiliaries and resistance heating)
<i>HP,total</i>	all heat pumps combined
<i>in</i>	incoming (with respect to the heat pump system)



<i>HEX _CE</i>	heat exchanger between condenser and evaporator
<i>is</i>	isentropic change of state
<i>l</i>	liquid
<i>max</i>	maximum
<i>mech</i>	mechanical (shaft)
<i>min</i>	minimum
<i>n</i>	night
<i>nom</i>	at nominal conditions
<i>out</i>	outgoing (with respect to the heat pump system)
<i>outdoor, in</i>	outdoor air entering the outdoor heat exchanger
<i>outdoor, out</i>	outdoor air leaving the outdoor heat exchanger
<i>prim</i>	primary energy
<i>pumps</i>	pumps for pumping water through the underground heat exchanger or from the underground aquifer
<i>rect</i>	rectifier
<i>R</i>	refrigerant
<i>solution</i>	ammonia-water solution
<i>sat</i>	saturation conditions
<i>start – up</i>	heat pump start-up
<i>sol _ pump</i>	solution pump
<i>strong</i>	strong solution
<i>sub</i>	subcooling
<i>sys</i>	entire system (including driving system, heat pump, auxiliaries and resistance heating)
<i>s, in</i>	entering the buffer vessel
<i>s, out</i>	leaving the buffer vessel
<i>SH</i>	space heating
<i>tot, yearly</i>	total, yearly
<i>v</i>	vapour
<i>vl</i>	vapour-liquid
<i>w</i>	water
<i>weak</i>	weak solution
<i>WH</i>	water heating
<i>Superscripts</i>	
<i>C</i>	cooling mode
<i>H</i>	heating mode



Abbreviations

<i>COP</i>	Coefficient Of Performance [$\text{kW}_{\text{th}}/\text{kW}_{\text{e}}$]
<i>DCF</i>	Defrost correction factor [-]
<i>EER</i>	Energy Efficiency Ratio [$\text{kW}_{\text{th}}/\text{kW}_{\text{e}}$, $\text{Btu}_{\text{th}}/\text{Wh}_{\text{e}}$]
<i>ESEER</i>	European Seasonal Energy Efficiency [$\text{kW}_{\text{th}}/\text{kW}_{\text{e}}$]
<i>FR</i>	Flow Ratio [-]
<i>GUE</i>	Gas Utilization Efficiency [$\text{kW}_{\text{th}}/\text{kW}_{\text{th}}$]
<i>GWP</i>	Global Warming Potential [$\text{kg CO}_2\text{-eq}$]
<i>HSPF</i>	Heating Season Performance Factor [$\text{Btu}_{\text{th}}/\text{Wh}_{\text{e}}$]
<i>LHV</i>	Lower Heating Value (no condensation of flue gases) [kJ/kg]
<i>NTU</i>	Number of Transfer Units [-]
<i>PER</i>	Primary Energy Ratio [$\text{kWh}_{\text{th}}/\text{kWh}_{\text{th}}$]
<i>RH</i>	Relative humidity [%]
<i>SEER</i>	Seasonal Energy Efficiency Ratio [$\text{Btu}_{\text{th}}/\text{Wh}_{\text{e}}$]
<i>SPF</i>	Seasonal Performance Factor [$\text{kWh}_{\text{th}}/\text{kWh}_{\text{e}}$, $\text{kWh}_{\text{th}}/\text{kWh}_{\text{th}}$]
<i>GAHP</i>	Gas Absorption Heat Pump
<i>AR</i>	reversible, air source
<i>GS</i>	ground source
<i>WS</i>	water source



1 Assignment description

In this chapter the goal and scope of the project are defined. It starts with the problem definition. After that it defines the problem that is encountered when applying heat pump systems for office buildings to buildings that have a different heat-to-cold ratio. The chapter further serves to explain the choices made and to place the assignment into perspective. It ends with a summary in the form of a flow diagram (see Figure 1 on page 3).

1.1 Problem definition

Following their success in the office building market, electrical heat pumps are considered more and more often to heat other large buildings, for example nursing homes. However, these applications have very different needs from office buildings. The most important differences lie in the fact that these buildings need much more heating than cooling, partly because of their construction but also partly because of the needs of their users. Therefore a mismatch exists between the amount of cold produced at the electrical heat pump's evaporator and the amount of cold required by these buildings. This is a problem when the electrical heat pump is used in combination with geothermal energy storage in underground aquifers, because regulations state that there has to be a match between the amount of heat and the amount of cold stored in these aquifers. Another problem that is commonly encountered when the heating systems of older buildings are renovated is that they commonly employ heat delivery systems that require high water supply temperatures in the range of 70 °C. These temperatures cannot be delivered at a reasonable efficiency by electrical heat pump systems.

The solution to the heat/cold mismatch would be to use a heat pump with a higher heat-to-cold ratio than the electrical heat pump systems. If this ratio matches better with the buildings own heat to cold ratio, geothermal energy storage can still be an attractive and feasible option for these applications. This could mean a large decrease in the energy consumption of such buildings. Another advantage would be that the size of the aquifer could be limited when high heat-to-cold ratio heat pumps are used, which would greatly reduce the investment costs of such projects.

Gas-fired heat pumps could provide this match. Both gas absorption heat pumps and gas engine driven heat pumps have a much higher heat to cold ratio than regular, electricity driven vapour-compression heat pumps. Another advantage of gas-fired heat pumps is that they are able to produce water of relatively high temperatures (~70 °C) according to their manufacturers. This would also solve the high supply temperature problem encountered in renovation projects.

1.2 Goal

The goal of the present project is to find out whether gas-fired heat pumps are a good means to lower the energy use for heating and cooling of buildings with a high ratio of heating demand to cooling demand.



Examples of such buildings are nursing homes. The comparison should not be limited to the low temperature heat which heat pumps normally provide; comparison at high water temperatures ($\sim 70^\circ\text{C}$) should also be included.

1.3 Scope

Since the goal of the project is defined in very general terms, one of the challenges was to narrow it down sufficiently in order to fit it in the three month period of this internship. For this reason the choice was made to focus exclusively on gas absorption heat pumps. The reason for this choice is that there is not too much experience with these types of machines within the Royal Haskoning company. More information and estimations on its performance are therefore more than welcome.

1.4 Approach

General market overview

Before going into the details of one specific type of heat pump, first the heat pump market is explored. In this general overview we zoom in on the gas absorption heat pumps.

Detailed heat pump modelling

In order to gain more understanding of the gas absorption heat pump, the reversible air source version of a commonly encountered gas absorption heat pump (Robur GAHP-AR) is modelled in detail. Another reason for this modelling effort is to check whether the manufacturer claims on its efficiency are reasonable.

Heating system modelling

In practice a heat pump is never on its own responsible for heating a building. Heating systems commonly consist of heat pumps which provide the base heating load, while gas-fired boilers are used to supply extra heating when peaks in the building heat demand occur. Because Royal Haskoning is interested in using gas absorption heat pumps in buildings, a heating system model was built. This model uses manufacturer data for gas absorption heat pumps using air, soil and water as heat sources to calculate the yearly energy use of the entire heating system.

Matching heat pump to application

Using the heat pump system model simulations for a reference climate year are performed. Using these simulations the energy use of the entire heating system is calculated. The amounts of heat and cold produced by the heat pump are monitored as well. The results of these simulations are used to check whether there is a good match between the amounts of heat and cold delivered by the heat pump and the amounts demanded by the building.

The whole approach is described above is summarized in Figure 1.



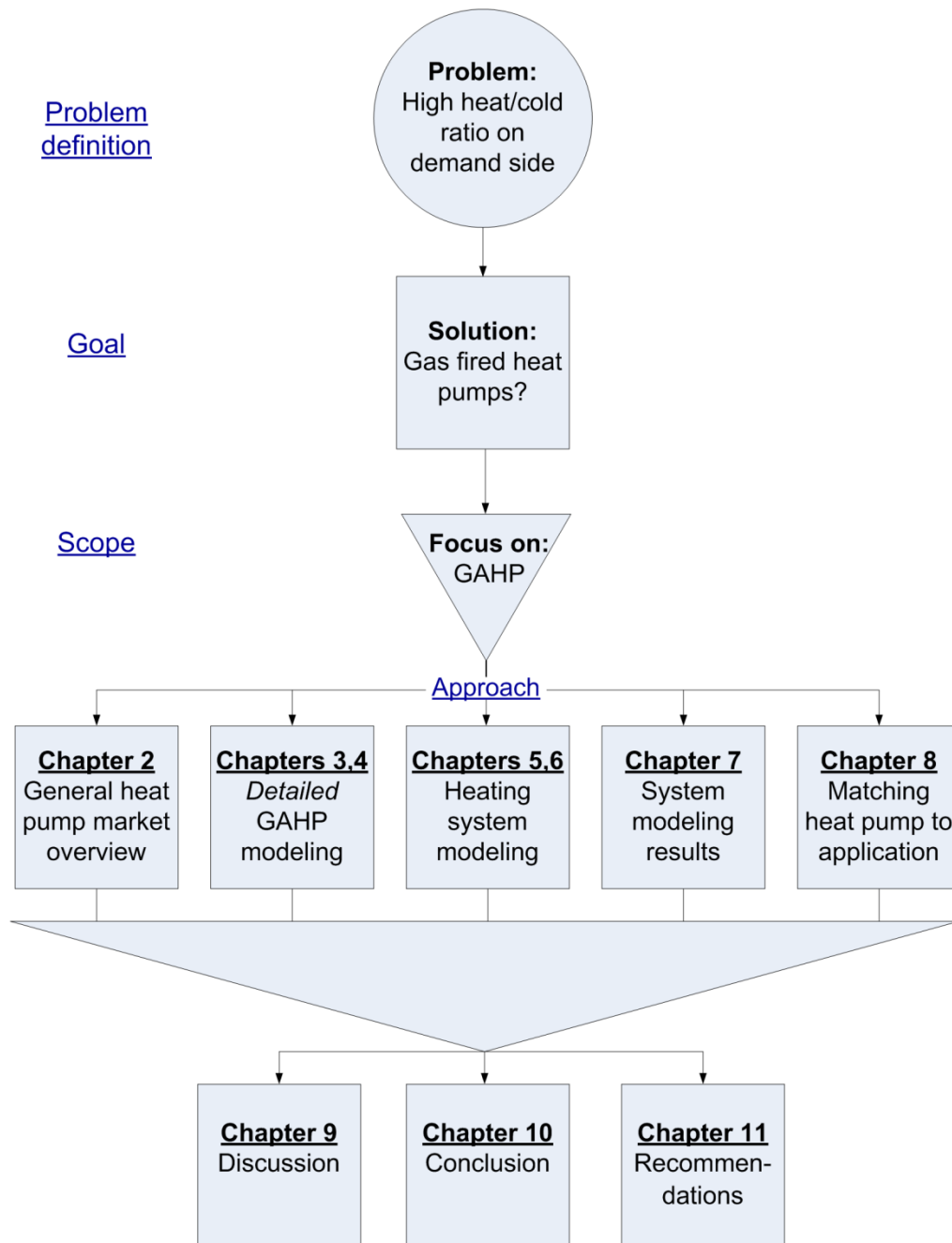


Figure 1. Graphical report overview. The overview shows how the problem definition is translated through the goal and the scope of the assignment into report chapters. GAHP = Gas Absorption Heat Pump.



2 General overview – Heat pumps for the built environment

This chapter provides a general overview of the different types of heat pumps used in the built environment. In the first section the underlying thermodynamic principles are discussed. The second section is about the classification of the different heat pump types according to working principle, driving energy source and heat source and heat sink. Section 1.3 deals with the different definitions that exist for the efficiency of a gas absorption heat pump. The fourth section lists the most important heat pump manufacturers. In the fifth section the operating ranges of various heat pump types is discussed. Part-load behaviour and temperature ranges are elucidated on.

2.1 Thermodynamic principles

In this section the operating principles of three heat pump systems are described. First the classic vapour-compression heat pump receives some attention. After that the absorption heat pumps is considered.

2.1.1 Vapour-compression

The commonly used electrical heat pump makes use of the vapour-compression cycle to get the desired heating effect. It uses a compressor to get the required pressure difference. Because the system has two pressure levels the boiling point of the refrigerant differs depending on its location in the system. The system lay-out is shown in Figure 2.

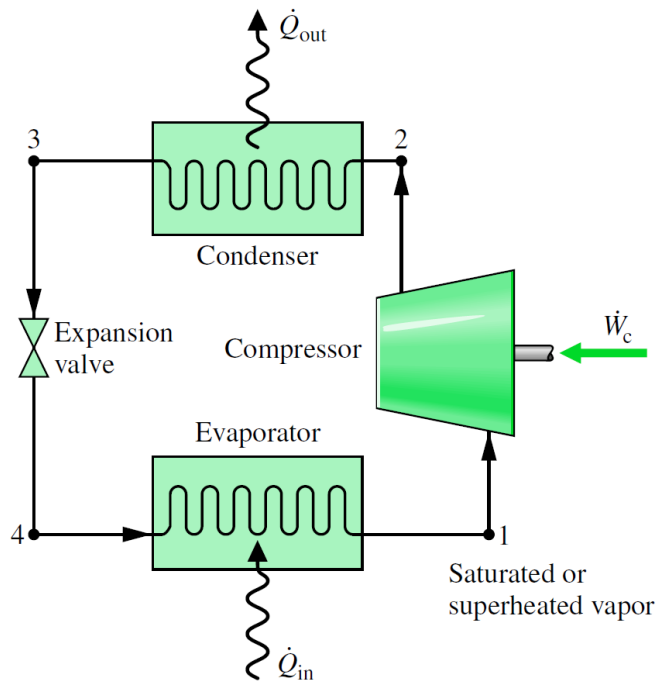


Figure 2. System layout for a vapour-compression heat pump. (Moran, 2006)

The arrows in Figure 1 indicate the direction in which the refrigerant flows. Starting at 1 the refrigerant vapour enters the compressor where it is compressed from the low pressure level of the evaporator to the high pressure level of the condenser. The (superheated) compressed refrigerant vapour then enters the condenser (point 2). In this heat exchanger it condenses, releasing heat to the building ($\dot{Q}_{out}$). The now liquid refrigerant leaves the condenser and flows through the expansion valve (point 3). Here its pressure is lowered back to evaporator pressure, resulting in a two-phase vapour-liquid mixture leaving the expansion valve (point 4). This mixture flows to the evaporator. Here heat from the heat source is used to evaporate the refrigerant ($\dot{Q}_{in}$). The resulting gaseous refrigerant is sucked into the compressor, repeating the cycle.

The vapour-compression cycle is often conveniently shown in a pressure-enthalpy diagram. As an example an ideal vapour compression cycle is shown in the pressure-enthalpy diagram for R134a in Figure 3. The evaporation takes place at $-3\text{ }^{\circ}\text{C}$ and condensation at $45\text{ }^{\circ}\text{C}$. The numbers correspond to the numbers for the different states used in Figure 2.

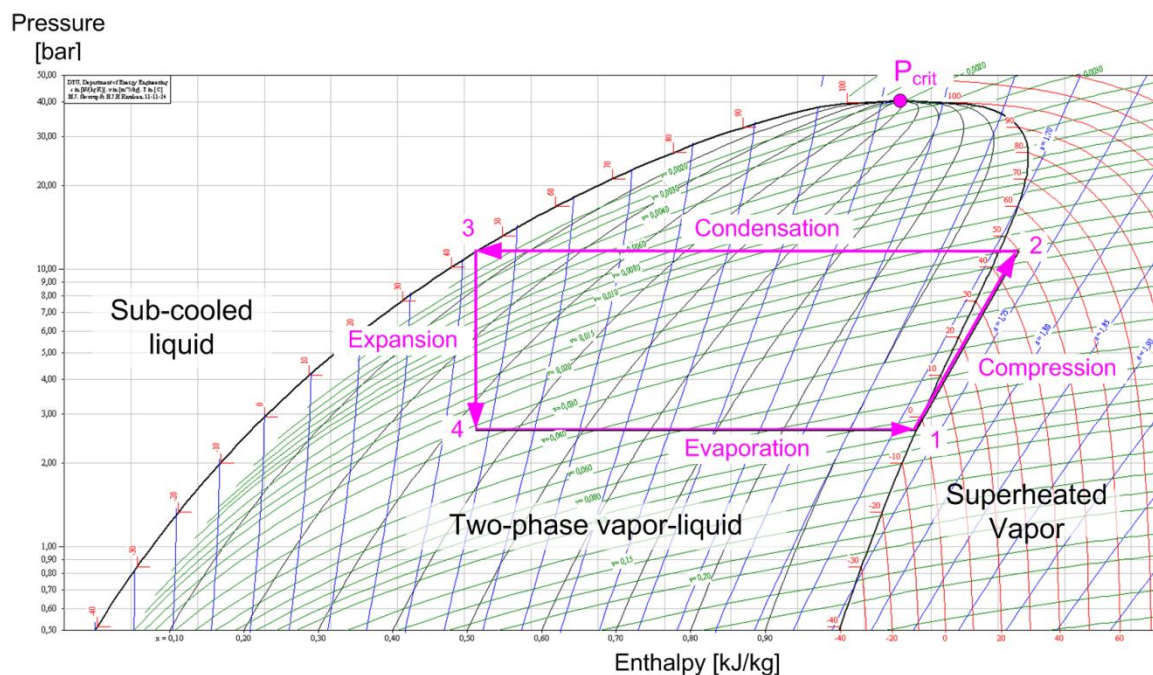


Figure 3. Pressure-enthalpy diagram for a vapour-compression cycle using R134a as the refrigerant. Evaporation takes place at $-3\text{ }^{\circ}\text{C}$, condensation at $45\text{ }^{\circ}\text{C}$. The considered cycle is ideal and the compression isentropic. No subcooling or superheating occurs.

In Figure 3 the thick black line separates the two phase area from the gas and the liquid phases. The critical pressure is indicated by a purple dot in the figure. The heat supplied to the evaporator to evaporate the refrigerant can be delivered by different sources. A common heat source is water from an underground aquifer. Another possibility is extracting the required heat from the air. A

third option involves taking the heat from the ground by using a heat exchanger below the ground surface.

A commonly quoted efficiency definition for heat pump systems is the coefficient of performance. Referring to Figure 2 it is defined as:

$$COP_{HP} = \frac{\dot{Q}_{out}}{\dot{W}_c} \quad (0.0.1)$$

There is a variety of fluids that can serve as refrigerants for vapour-compression cycles. Whether a certain fluid is suitable to be a refrigerant depends on its properties. More information on refrigerants and their properties is given in Appendix H.

2.1.2 Absorption

Absorption principle

The principle of the absorption heat pump is similar to the principle of the vapour-compression heat pump in many respects. It has both an evaporator to take heat from the heat source and a condenser to transfer the heat from the refrigerant to the building.

The main difference between both systems is that the absorption heat pump does not have a compressor to increase the pressure level of the refrigerant. Instead it uses an intricate system consisting of a pump, a generator and an absorber. The system layout is shown in Figure 4. It can be shown in this figure that the compressor of Figure 2 has been replaced by a more complex system.

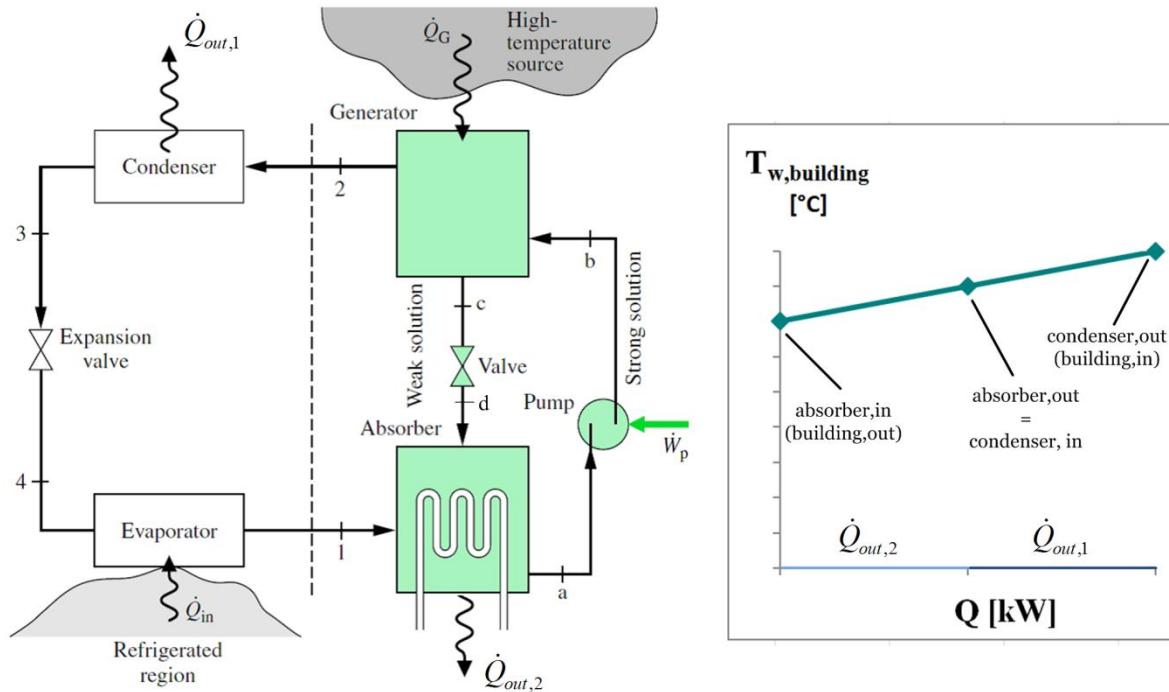


Figure 4. Simplified ammonia-water heat pump system. In this simplified drawing the internal heat exchanger is not shown (Moran, 2006). The diagram on the right shows the temperature of the building heating water as a function of the amount of heat supplied.

In Figure 4 the refrigerant is ammonia. Starting at 1 the ammonia vapour coming from the evaporator is led to the absorber. There it is mixed with the solution labelled 'c' in Figure 4. This 'weak solution' consists of water with some dissolved ammonia. When this solution is mixed with the pure ammonia vapour from the evaporator, heat is released which can be utilized to heat up the building heating water ($\dot{Q}_{out,2}$). The resulting solution (stream a), which is now 'strong' in ammonia, is pumped to a higher pressure level. Then the solution enters the generator. In the generator heat is supplied to the solution to force the ammonia vapour out again. It is this heat that drives the cycle, together with the electric power used by the pump. The vapour that comes out of the generator enters the condenser, after which the cycle is similar to the vapour-compression cycle again. The condenser heat ($\dot{Q}_{out,1}$) is used to heat the building heating water even further. In the generator a weak solution results when the ammonia vapour is forced out. This is returned back to the absorber to repeat the absorption process.

The ammonia-water cycle of Figure 4 can be conveniently displayed in a pressure-temperature diagram showing lines of constant ammonia concentration. This is done in Figure 5.

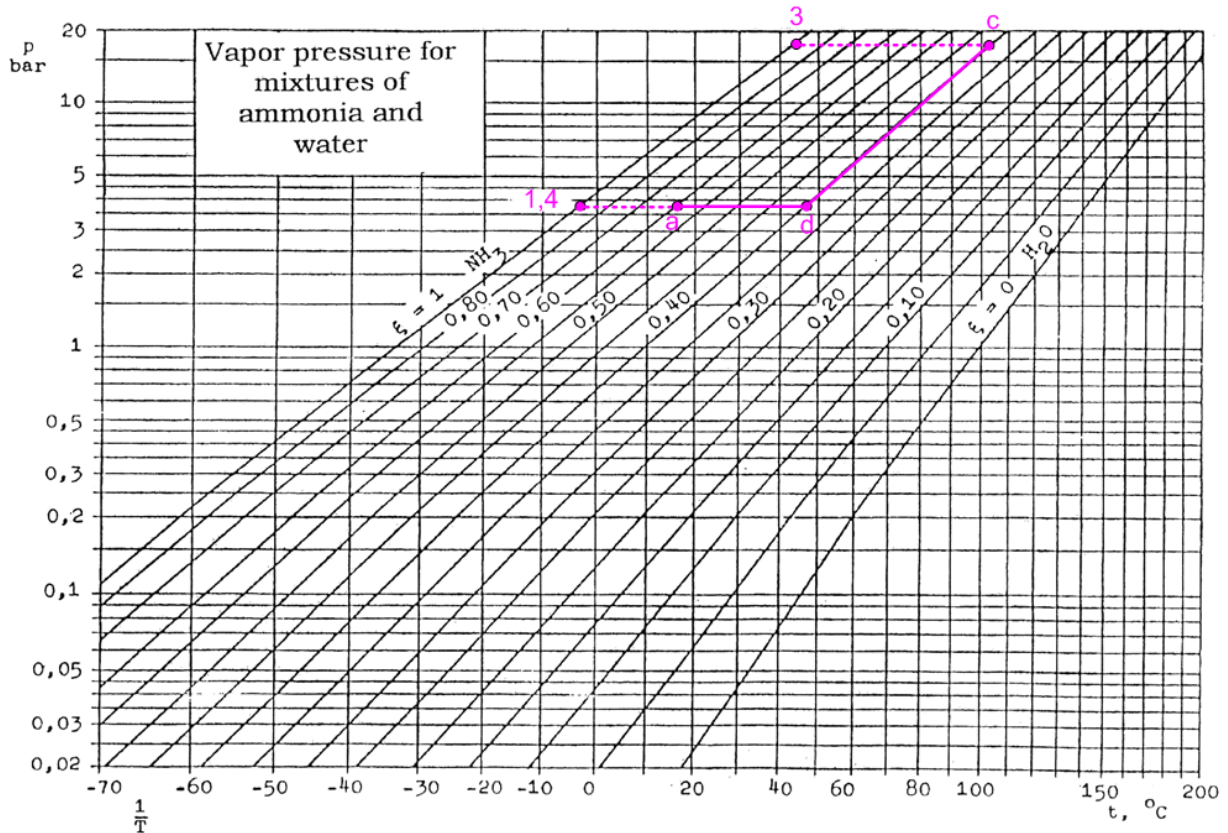


Figure 5. Pressure temperature diagram for the ammonia water cycle. (Podesser, 2003)

In Figure 5 the saturation pressure is on the vertical axis and the saturation temperature on the horizontal axis. It is assumed that the absorber and the evaporator are at the same pressure, as are the generator and the condenser. The weak solution is taken to have an ammonia mass fraction of 0.4, the strong solution has a mass fraction of ammonia of 0.6. Since the diagram only shows saturation conditions, subcooled or superheated streams from Figure 4 are not shown in Figure 5.

Ammonia-water systems

It should be noted that the cycle shown in Figure 4 is simplified. In real ammonia-water systems the ammonia vapour leaving the generator is not pure. It therefore has to be purified in a so-called rectifier before entering the condenser. Furthermore there is often a heat exchanger between generator and absorber to exchange heat between streams **a** and **c**. This last measure decreases the amount of heat needed to force the ammonia vapour out in the generator and therefore it increases the COP of the cycle. Still more complicated systems exist, with a lot of internal heat recovery and so-called generator-absorber heat exchange (GAX). In this last type of system the absorption process is split into two parts, making it possible for the absorption and the generator to have some temperature overlap. Direct heat exchange between both processes further increases energy efficiency. The ammonia-water absorption heat pump modelled in Chapter 3 works according to this GAX principle.

LiBr-water systems

There are also systems available that use LiBr-water in an absorption cycle. In that case water acts as the refrigerant. Due to the physical properties of the LiBr-water mixture, the COP values for these systems are higher than those obtained from an ammonia cycle using the same condenser and evaporator temperatures. Because water is the refrigerant, LiBr-water systems cannot have very low evaporator temperatures. In practice the minimum achievable evaporator temperature is about 4 °C. For this reason LiBr-water heat pumps cannot use outside air as a heat source. Aquifers or ground water provide a good alternative. Although LiBr-water systems are well-developed and often encountered in industrial applications, no examples of the use of LiBr-water heat pumps systems for the built environment were found.

2.1.3 Adsorption

Next to the absorption cycle there also exists an adsorption cycle. In this cycle the refrigerant is adsorbed onto a solid adsorbent, instead of being absorbed into a liquid absorbent. The interested reader can find more about the underlying principles of the adsorption cycle in Appendix B.

2.2 Heat pump classification in types

Heat pumps can be classified in numerous ways. From a technical point of view a logical classification is according to working principle. When this is done only two *commonly used* types remain: the vapour-compression heat pump and the absorption heat pump. These types can be subdivided further according to the heat source and heat sink used. The heat source is the medium from which the heat pump obtains its low temperature heat, while the heat sink is the medium to which the heat pump transfers its higher temperature heat. Other differences include the energy source used to drive the heat pump. This can be natural gas, electricity or waste heat. An overview of the most commonly used heat pumps and their sub-classification is shown in Figure 6.

In Figure 6 there are two gas-fired heat pumps: a vapour-compression heat pump driven by a gas engine and an absorption heat pump driven by the heat from natural gas. In case the gas powered heat pump uses a gas engine to drive the compressor of a vapour-compression cycle it is called a *gas engine heat pump*. When an absorption heat pump uses natural gas to provide the heat needed to drive it the system is called a *gas absorption heat pump*.



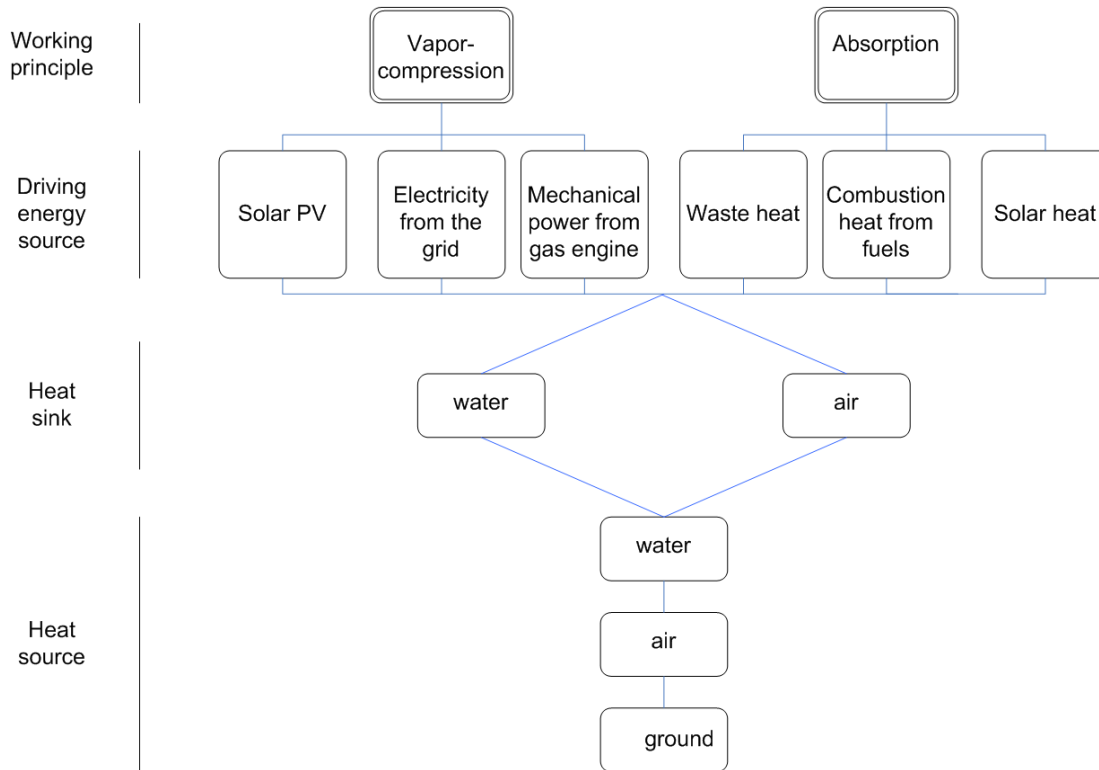


Figure 6. Classification of the *most commonly used* heat pumps.

2.3 Efficiency definitions – gas absorption heat pump

There are a lot of different ways to assess heat pump performance. For gas absorption heat pumps the efficiency definitions are given in this section. The efficiency definitions for electric heat pumps and gas engine powered heat pumps are given in Appendix C.

2.3.1 Gas Utilization Efficiency

Gas absorption heat pumps do not use the COP to measure their performance because they do not use electricity to power them. For these heat pumps the *Gas Utilization Efficiency (GUE)* is used. The PER is, by definition, equal to the GUE. For heating mode it is defined as:

$$GUE_{HP,heating} = PER_{HP,heating} = \frac{\dot{Q}_{cond} + \dot{Q}_{abs}}{\dot{m}_{gas,HP} LHV_{gas}} \quad (0.0.2)$$

In the denominator the amount of heat added by burning natural gas is represented. The definition for the *GUE* in cooling mode is completely analogous to equation (0.0.2) only then the term $(\dot{Q}_{cond} + \dot{Q}_{abs})$ is replaced by $\dot{Q}_{evap}$. Note that absorption heat pumps do not only use the heat from the condenser but also the heat from the absorber for water heating.

2.3.2 Seasonal performance factor

The primary seasonal performance factor for the heating season equals the primary energy ratio of the entire heating system. In this definition auxiliary electricity demand is taken into account, as is the gas use for supplementary heating by a gas-fired boiler. The gas-fired boiler is found in this efficiency definition because heating systems seldom only include heat pumps. The seasonal performance factor is calculated as:

$$SPF_{sys} = SPF_{prim} = \frac{\int (\dot{Q}_{WH} + \dot{Q}_{SH}) \cdot dt}{\int (\dot{m}_{gas,HP} LHV_{gas} + \dot{E}_{aux} / \eta_{el,grid} + \dot{m}_{gas,boiler} LHV_{gas}) \cdot dt} \quad (0.0.3)$$

Similar definitions exist for rating the cooling performance.

2.4 Market overview – Available systems & applications

A wide range of heat pumps is available on the market today. The first heat pump systems to be used on a commercial scale in the built environment used the vapour-compression cycle to get the desired heating effect. Recently the gas-powered heat pumps have entered the market. These can be either gas engine heat pumps or gas absorption heat pumps.

As noted above the most important heat pumps available on a commercial scale are the vapour-compression and the absorption heat pump. The vapour-compression heat pump can be either gas- or electricity driven, while the absorption heat pump is either gas or waste heat driven.

Some important manufacturers for vapour-compression heat pumps for the built environment are:

- Trane (US)
- Carrier (US)
- Climaveneta (Italy)
- Ciat (France)
- York (US)
- Daikin (Japan)

For absorption heat pumps there are also a number of large manufacturers, most of them produce heat pumps for industrial applications:

- Carrier (US) - LiBr-Water
- Hitachi (Japan) - LiBr-Water
- Thermax (India) - LiBr-Water, Ammonia-Water
- York (US) - LiBr-Water
- Colibri (NL) - Ammonia-Water
- Robur (Italy) - Ammonia-Water

Two well-known gas engine driven heat pump manufacturers are Aisin-Toyota and Sanyo. Both companies are Japanese. Capacities are about 80 kW for heating and 60 kW for cooling, depending on the gas engine used. Units are placed in parallel if larger capacities are required. Part-load is also possible. This is done by reducing the engine rotational speed at constant torque. Multiple compressors are driven by the same engine, making it possible to switch off one or more compressors during part-load operation. There is also a Dutch company that produces gas engine driven heat pumps. The company is called Reduses and it produces gas engine heat pumps of reasonable sizes, ranging from 137 to 280 kW for low temperature heating and from 57 to 99 kW for high temperature heating. Stepless part-load operation from 25 to 100 % is possible for these systems (Reduses, 2012).

The market for the gas absorption heat pumps in the built environment is dominated by the Italian company Robur. Its heat pumps are sold in Europe under the names Robur, Buderus and Remeha. The capacities of a single unit go up to about 40 kW, depending on the heat source used (Techneco, 2011b). Larger capacities are obtained by parallel operation. 'Part-load operation' is accomplished by switching off one or more units. A modulating unit can also be purchased, but the required controller is expensive and therefore this is seldom done in practice.

Robur's absorption heat pumps use the ammonia-water cycle. Because of the relatively large amounts of ammonia and water inside, the system needs about 15 to 20 minutes to start-up. Furthermore the amounts of starts and stops should be limited. Each unit is supplied with a buffer vessel to limit the amounts of starts and stops and to store surplus thermal energy. When more units are in parallel, they can use the same buffer vessel.

2.5 Heat pump properties – market claims

As explained in the previous section, there are quite a number of different heat pumps available on the market. Each heat pump has its own characteristic when it comes to temperatures that can be delivered and part-load behaviour. In this section these characteristics are discussed for electrical heat pumps, gas engine driven heat pumps and gas absorption heat pumps.

2.5.1 Temperature range

The supply temperature to the building a certain heat pump can deliver depends on the heat pump type. The following general temperature ranges are valid for the various types of heating systems (Figure 7).



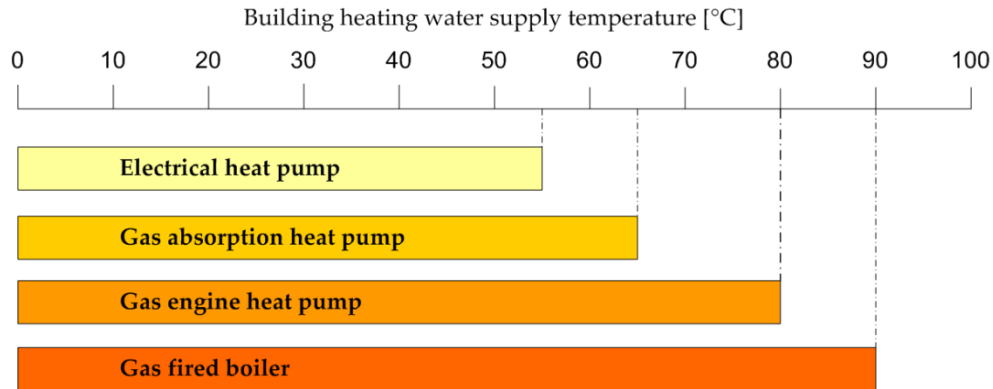


Figure 7. Temperature ranges for various heating systems available on the market.

It can be seen from Figure 7 that gas-fired systems are able to reach higher temperatures than electrical heat pumps. The reason that the gas engine driven heat pump can reach temperatures of 80 °C is that it utilizes high temperature waste heat from the gas engine to heat up the water.

2.5.2 Part-load possibilities

The different heat pumps also differ in the load range they are able to cope with. In general absorption heat pumps are less well capable of part-load than vapour compression heat pumps. This can also be seen from Table 1.

Table 1. Load range attainable with the various heat pump types.

	Load range [%]
Electrical vapour-compression heat pumps	20-100
Gas engine driven heat pumps	25-100
Gas absorption heat pumps	50-100

3 Modelling approach – Detailed GAX absorption heat pump model

In the previous chapter we have seen that there are different types of gas-fired heat pumps. As described in Chapter 1 the focus is on gas absorption heat pumps. The gas absorption heat pump market for buildings is small and there is only one manufacturer of these machines: Robur. Since this machine is not yet monitored by Royal Haskoning, no field data on its performance is available. The only efficiency data for the machine is coming from Robur itself. Since manufacturers often perform their measurements in such a way that their machines will show good performance in the tests, this data cannot be trusted without testing it. To test the data a thermodynamic model of one of Robur's gas absorption heat pumps is built. The heat pump makes use of the ammonia-water GAX absorption cycle. The present chapter describes the modelling approach followed. The first section describes the model's inputs and outputs. In the second section the modelling is described in more detail.

3.1 Model input and output

The modelling of the Robur GAHP-AR heat pump is described in this chapter. This heat pump is a reversible gas absorption heat pump using air as the heat source. The heat pump cannot operate under part-load, which means that its gas burner heating power is constant. Its nominal heating capacity is about 35 kW^1 , while its cooling capacity is roughly 17 kW^2 . The model of the heat pump has to predict its heating power with varying building inlet and outlet temperatures and for varying outdoor air temperatures. This is shown in Figure 8.

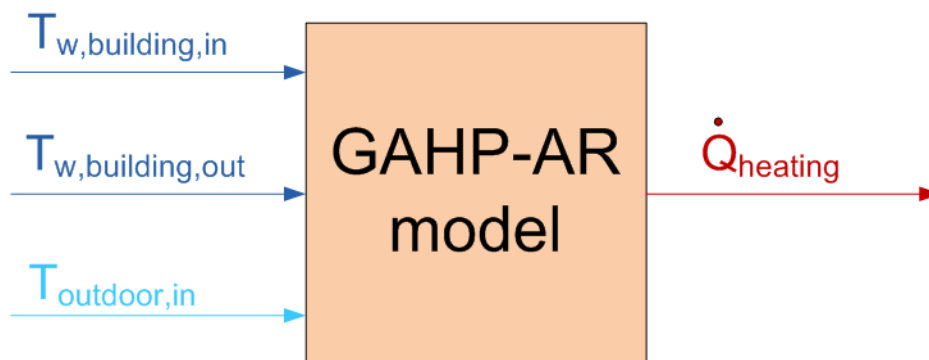


Figure 8. Simplified representation of the model showing its inputs and outputs.

3.2 Absorption heat pump model

The model of the absorption heat pump consists of several modules which are programmed in Matlab/Simulink. The modules are made such that they can be used for other models as well. In the

¹ The nominal conditions for heating are an outside air temperature of 7°C and an outlet temperature of 50°C for the heated water.

² For cooling nominal conditions are an outdoor air temperature of 35°C and a cooled water temperature of 7°C .

case of the evaporator, the condenser and the expansion valve this means that they can be used for modelling a vapour-compression heat pump or cooling machine if a compressor module is added to the system. The model relies on thermodynamic properties for refrigerant, water and air from the libraries of FluidProp. FluidProp is a thermodynamic software tool that is able to calculate thermodynamic properties of a wide variety of industrial fluids. By choosing a different fluid from the FluidProp library, the heat pump cycle can be operated with a different refrigerant.

3.2.1 Information on the Robur GAHP-AR GAX absorption heat pump

The system layout for the Robur GAHP-AR ammonia-water heat pump is displayed in Figure 9. See also section 2.1.2 on page 7 for a basic explanation of the underlying principles of the ammonia-water absorption cycle.

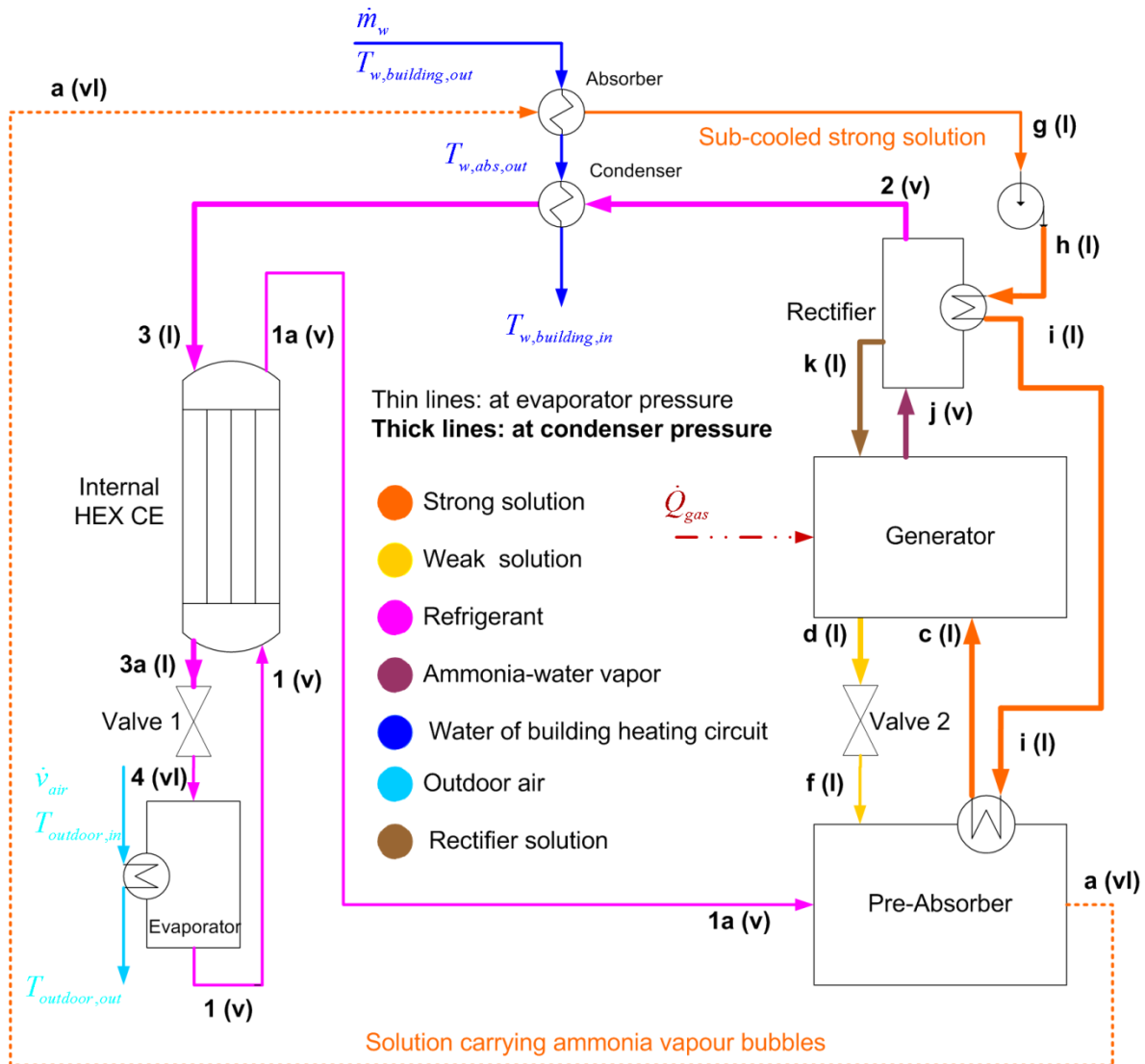


Figure 9. Simplified process flow diagram of the Robur GAHP-AR air source reversible gas-fired heat pump in heating mode. The stream labelling is used for the system modelling. v = vapour, l = liquid, vl = vapour-liquid.

It is instructive to follow the path of the refrigerant, which is ammonia in this case, through the heat pump of Figure 9 in order to explain the functions of the different components.

With reference to Figure 9 we start at 1a when the ammonia vapour enters the pre-absorber, where it is brought into contact with the weak solution from the generator (stream f). This weak solution consists of a liquid mixture of ammonia and water with a relatively low ammonia concentration. Upon mixing of streams 1a and f absorption of gaseous ammonia into the liquid phase takes place, releasing a large amount of heat. This heat is taken away by heat exchange with stream i, which will be discussed later. The solution leaving the pre-absorber (stream a) has a temperature higher than its saturation temperature. It therefore consists of an ammonia-water solution which carries entrained ammonia bubbles that cannot be absorbed at these conditions.

Stream a is led to the absorber, where it is used to pre-heat the water coming from the building. While stream a is cooled towards its saturation temperature, the amount of absorbed ammonia increases, releasing a lot of heat. A sub-cooled ammonia-water solution leaves the absorber to enter the solution pump (stream g).

The solution pump then brings the strong solution (stream g) from evaporator pressure to condenser pressure. The stream leaving the solution pump (stream h) is now used to cool the rectifier and the pre-absorber. When it enters the generator (stream c) it is already substantially pre-heated, lowering the generator duty and therefore its gas use. This had a positive effect on the overall system efficiency.

The function of the generator is to produce ammonia refrigerant. This is done by adding heat to the entering strong solution by means of a gas burner, which is installed at the lower section of the generator. Upon heating up the strong solution above its saturation temperature bubbles of ammonia and water vapour are released until the concentration in the solution stabilizes at the equilibrium value corresponding to local temperature and pressure levels in the generator. The resulting (weak) solution (stream d) has a lower ammonia concentration than the entering (strong) solution and it is returned to the pre-absorber through an expansion valve to lower its pressure back to evaporator pressure (stream f). The two-component vapour mixture which is produced in the generator is purified as it rises across the trays, further increasing its ammonia concentration. The enriched vapour phase (stream j) is then passed to the rectifier. The vapour is cooled and partially condensed here. This is done to condense out the remaining water from the enriched vapour stream. Unfortunately this also means condensing out some ammonia, which is undesirable but cannot be avoided. A reflux stream k is led back to the generator, while the now pure ammonia vapour flows to the condenser (stream 2).

In the condenser the ammonia vapour condenses, releasing heat to the water already pre-heated by the absorber. After the condenser the water should be at the temperature level desired by the



building heating system ($T_{w,building,in}$). Ammonia leaves the condenser as a saturated liquid (stream **3**). It then passes through the internal heat exchanger, where it loses heat to the vapour coming from the evaporator (stream **1**).

After expansion through a valve, the resulting two-phase ammonia stream enters the evaporator (stream **4**). A large fan blows outdoor air past the evaporator coils. Inside the coils the part of the ammonia that is still in the liquid phase evaporates, using heat from the outdoor air. It leaves as saturated vapour (stream **1**).

In the internal heat exchanger the vapour from the evaporator is heated by the liquid ammonia from the condenser. The superheated ammonia vapour (stream **1a**) now enters the pre-absorber after which the whole cycle repeats itself.

In case the heat pump is used in cooling mode, the condensation and absorption steps take place in the evaporator coils, while the evaporation of the refrigerant takes place in the condenser/absorber component to cool the water instead of heating it. The heat pump in cooling mode is not considered here.

3.2.2 Modelling approach

Modelling always involves making assumptions. This model is no exception. Before discussing the assumptions of each component first some assumptions are given which are generally valid for the heat pump model.

General modelling assumptions

- No pressure drop inside any of the components (except the expansion valves)
- No pressure drop in the connecting lines
- Uniform pressure inside each of the components
- Fixed water supply temperature to the absorber in heating mode
- Steady state (no storage of mass, momentum or energy)

The assumptions for each separate component are shown in Figure 10 and Figure 11. These figures provide a very general picture. Further details of the modelling process such as the equations and parameter values used for all components can be read in Appendix G.



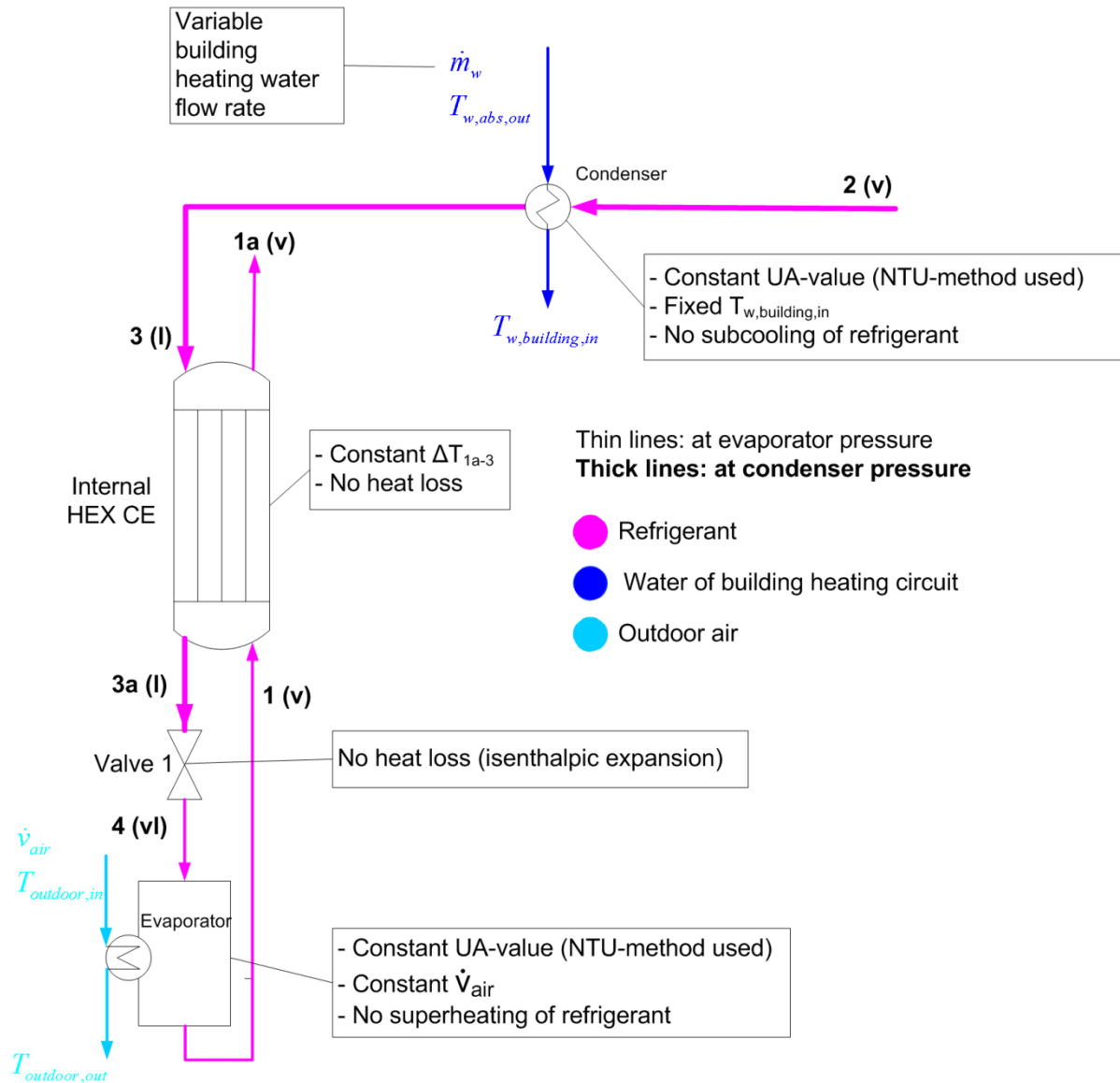


Figure 10. Part 1 of the simplified process flow diagram of the Robur GAHP-AR air source reversible gas-fired heat pump in heating mode. The stream labelling is used for the system modelling. v = vapour, l = liquid, vl = vapour-liquid. The process flow diagram shows the modelling assumptions. The NTU method is explained later in this section.

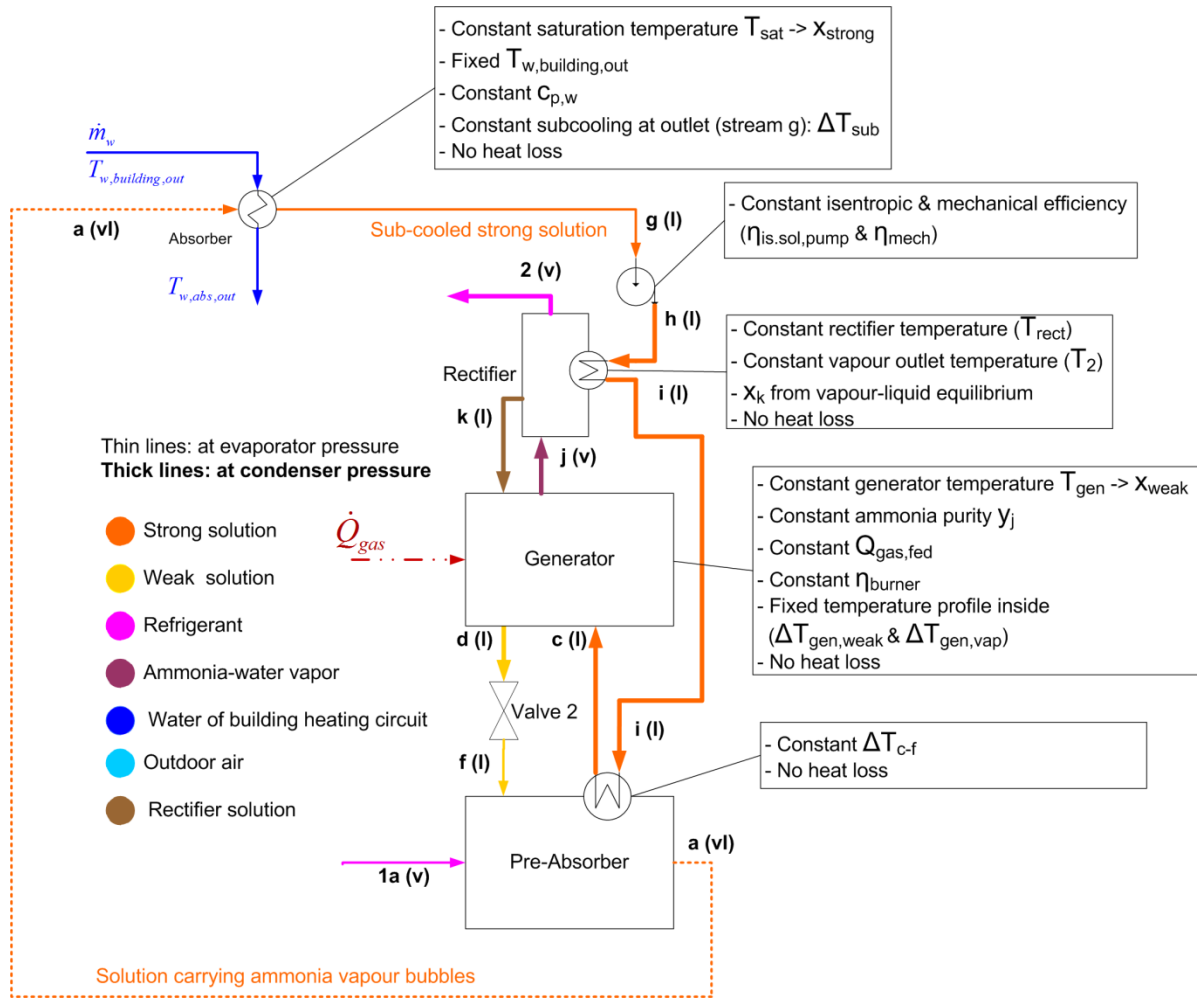


Figure 11. Part 2 of the simplified process flow diagram of the Robur GAHP-AR air source reversible gas-fired heat pump in heating mode. The stream labelling is used for the system modelling. v = vapour, l = liquid, vl = vapour-liquid. The process flow diagram shows the modelling assumptions. The NTU method is explained later in this section.

Modular model structure

In order to solve the system, each component is modelled as a separate module, with certain pre-defined inputs and outputs from a computational point of view. Care was taken to make sure that the number of outputs equals the number of independent equations. As an example, the model of the absorber component is explained below.

Model of the absorber component

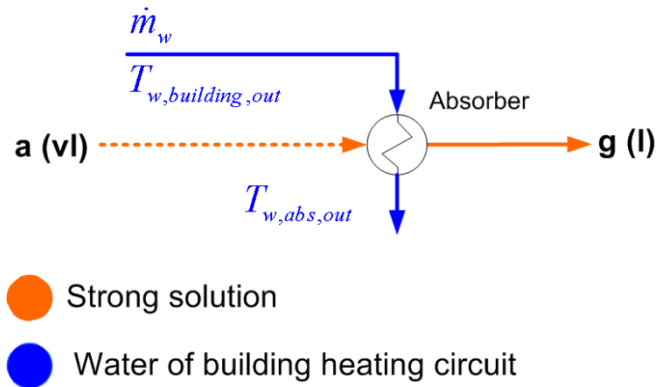


Figure 12. Modelling layout: Absorber.

Inputs

$$x_{strong}, T_{w,building,out} = 35^\circ\text{C}, \dot{m}_w, \dot{m}_{strong}, h_a, P_{evap}$$

Parameters

$$T_{sat} = 50^\circ\text{C}^{(1)}, c_{p,w} = 4.1797 \text{ kJ/kg} \cdot \text{K}^{(2)}$$

$$\Delta T_{sub} = 5 \text{ K}^{(3)}$$

Outputs (4)

$$T_g, h_g, \dot{Q}_{solution}, T_{w,abs,out}$$

Assumptions

- Fixed sub-cooling of the strong solution at the absorber outlet
- No heat loss (adiabatic absorber)
- Constant specific heat capacity for the building heating water at the given temperature range

Equations

In the absorber the building heating water is heated. The ammonia-water solution from the pre-absorber with ammonia bubbles inside is cooled down by the thermal contact with the building heating water. It is assumed that the ammonia-water solution leaves the absorber sub-cooled:

$$T_g = T_{sat} - \Delta T_{sub} \quad (3.3.1)$$

With the temperature known, the enthalpy of the strong solution leaving the absorber can be calculated:

$$h_g = h(x_{strong}, P_{evap}, T_g) \quad (3.3.2)$$

Since the enthalpy and mass flow rate of the incoming strong solution are known and mass conservation is valid for the absorber, the amount of heat released can be calculated:

$$\dot{Q}_{solution} = \dot{m}_{strong} (h_a - h_g) \quad (3.3.3)$$

The temperature of the building heating water leaving the absorber can now be calculated from the energy balance on the absorber:

$$T_{w,abs,out} = T_{w,building,out} + \frac{\dot{Q}_{solution}}{\dot{m}_w c_{p,w}} \quad (3.3.4)$$

The modules of the other components have a similar computational structure as the absorber. They are elucidated on in Appendix G. The modules of each of the components are coupled to each other in Simulink and since each component has zero degrees of freedom, the system can be solved.

Simulink modelling limitations

The heat pump performance is predicted using a Simulink model. Although Simulink is a great tool to use for system modelling, the program proved to have difficulties with more complicated models like this one with a lot of algebraic loops. To get the model running, initial guesses have to be provided for a number of system variables. These initial guesses have to be chosen carefully. Providing the wrong guesses results in a model that will not run at all. When the number of algebraic loops increases, the number of initial guesses that has to be provided increases as well. Because the initial guesses have to be consistent with respect to each other to make sure that the initial solution provided is a feasible one, choosing the right values gets increasingly difficult with the number of initial guesses increasing. It is for this reason that not all heat exchangers in the process were modelled using the NTU-method, since each time this method is used it adds another algebraic loop to the system. To bypass this problem, most heat exchangers were modelled by assuming a fixed temperature difference between certain outlet streams.



Explanation of the NTU-method

Only two heat exchangers were modelled using the NTU method: the condenser and the evaporator. This was done because the condensation and the evaporation temperature determine the two pressure levels in the system and those should be allowed to vary.

Using the NTU method for these heat exchangers is convenient, since the evaporation and condensation of the refrigerant takes place at constant temperature, which means the NTU-method can be simplified to the method for a so-called single-stream device. The effectiveness of a single stream heat exchanger is given by:

$$\varepsilon = 1 - \exp(-NTU) \quad (3.3.5)$$

with the number of transfer units (NTU) defined as:

$$NTU = \frac{(UA)}{\dot{m}_{medium} c_{p,medium}} \quad (3.3.6)$$

The medium is water for the condenser and outdoor air for the evaporator in this case. The heat exchanger effectiveness is defined as the actual heat transfer divided by the maximum heat transfer obtainable in an infinitely long exchanger. For the condenser we get:

$$\varepsilon_{condenser} = \frac{T_{w,building,in} - T_{w,abs,out}}{T_3 - T_{w,abs,out}} \quad (3.3.7)$$

And for the evaporator we have:

$$\varepsilon_{evaporator} = \frac{T_{outdoor,in} - T_{outdoor,out}}{T_{outdoor,in} - T_1} \quad (3.3.8)$$

Solution concentrations

The concentrations of the weak and the strong solution depend on the temperature and pressure in certain system components. The concentration of the weak solution is calculated in the generator from the condensation pressure and the generator temperature:

$$x_{weak} = x(P_{cond}, T_{gen}) \quad (3.3.9)$$

For the strong solution the concentration is set by the saturation temperature and the evaporating pressure:

$$x_{strong} = x(P_{evap}, T_{sat}) \quad (3.3.10)$$



Constant gas burner heating power

The gas absorption heat pump is driven by either LPG or natural gas. The gas burner is installed at the bottom of the generator. In the Robur GAHP-AR heat pump the hot flue gases from natural gas combustion first transfer a part of their heat to the walls of the generator, which are covered with heat transfer fins. The combustion gases then leave the system through its stack. The successor of the GAHP-AR (the Proline GAHP-AR) uses flue gas cooling and condensation to extract additional heat from the flue gases. This has a large positive effect on the overall system efficiency.

As stated earlier the GAHP-AR heat pump, which is modelled here, is not able to function under part-load conditions. Therefore the amount of natural gas fed is constant at its maximum value when the heat pump is in operation. In order to account for heat loss with the flue gases and radiation losses from the generator's insulation materials a constant burner efficiency is assumed. This means that the amount of heat that is added to the generator's energy balance is:

$$\dot{Q}_{gas} = \dot{Q}_{gas, fed} \eta_{burner}$$



4 Model validation – Detailed heat pump model

The heat pump model of Chapter 3 is validated in this chapter. Section 4.1 shows the results from the model for one set of operating conditions. In the second section of this chapter a qualitative validation is performed. The final section of this chapter is on the quantitative validation of the results by comparison to manufacturer data.

4.1 Modelling results – Reference case

The model holds a lot of information on the conditions at various locations inside the absorption heat pump modelled. In this section this information is shown for the reference case. The inputs and parameter values for the reference case are as described in Appendix G. They are summarized in Table 2.

Table 2. Inputs and parameter values for the heat pump model in the reference situation.

Inputs			
$T_{w,building,out}$ [°C]	35		
$T_{w,building,in}$ [°C]	45		
$T_{outdoor,in}$ [°C]	2		
Parameters			
UA_c [kW/K]	2.73	T_{sat} [°C]	50
UA_e [kW/K]	2.76	$\Delta T_{gen,vap}$ [K]	15
V_{air} [m ³ /s]	3.33	$\Delta T_{gen,weak}$ [K]	15
T_{gen} [°C]	150	$\Delta T_{rect,vap}$ [K]	20
T_{rect} [°C]	76	ΔT_{sub} [K]	5
y_j [kg NH ₃ /kg]	0.96	ΔT_{cf} [K]	10
$Q_{gas,fed}$ [kW]	25.7	ΔT_{1a-3} [K]	20
$\eta_{pump,is}$ [-]	0.5	P_{air} [bar]	1.01325
$\eta_{pump,mech}$ [-]	0.98	$c_{p,w}$ [kJ/kg·K]	4.1797
η_{burner} [-]	0.94		

The results of the calculation are shown in the process flow diagram in Figure 13.



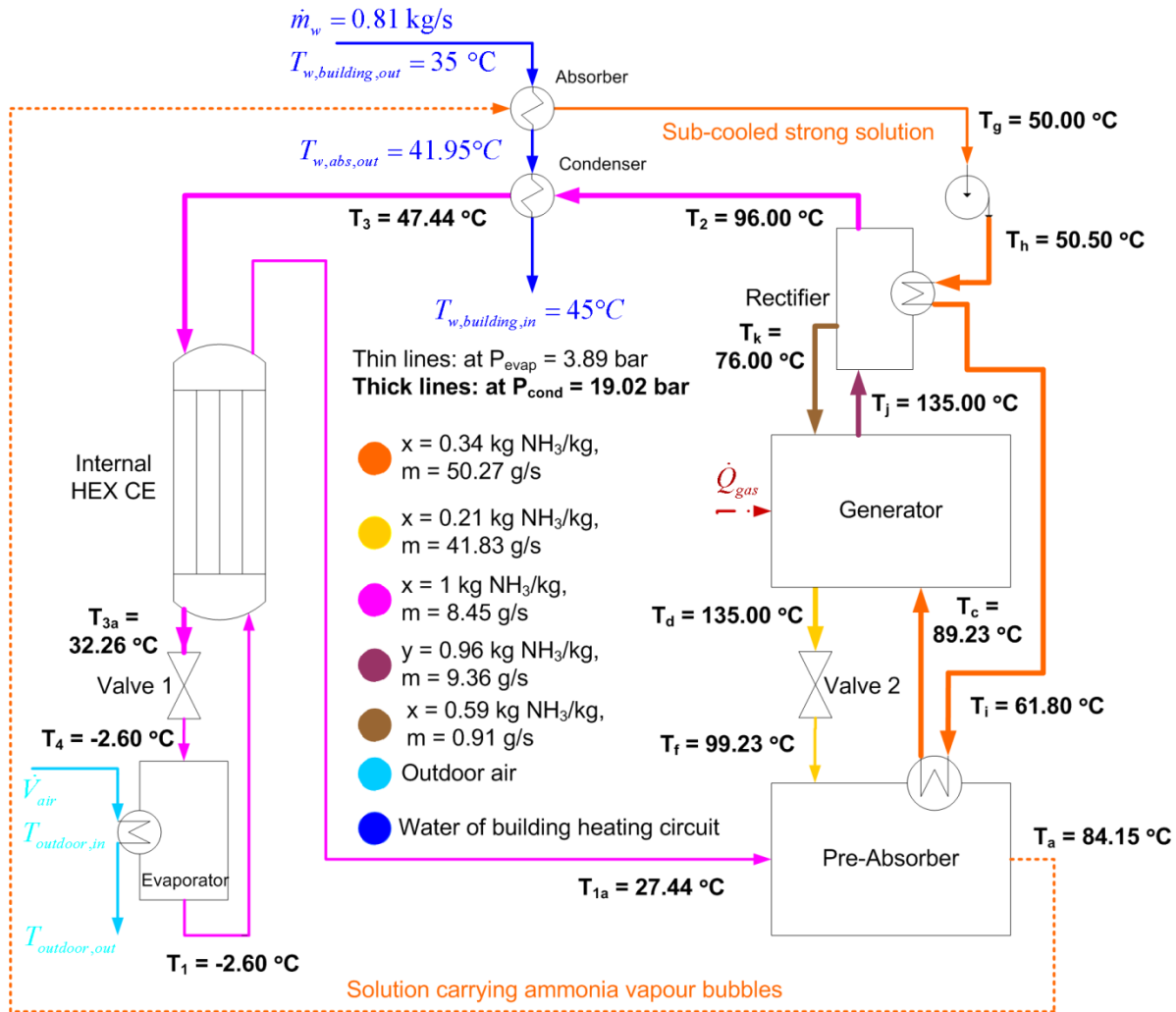


Figure 13. Process flow diagram showing the results of the calculation using the inputs and parameters from Table 2.

4.2 Qualitative validation

4.2.1 Sensitivity analysis

In order to see whether the heat pump model behaves as expected, a number of parameters are changed and their effect on the Gas Utilization Efficiency (GUE) reported. The definition of the GUE was explained in Chapter 1 and is reprinted below:

$$GUE_{\text{HP, heating}} = \frac{\dot{Q}_{\text{cond}} + \dot{Q}_{\text{abs}}}{\dot{m}_{\text{gas, HP}} LHV_{\text{gas}}} \quad (4.2.1)$$

In the simulations one (or two) parameters are varied, while the others are kept constant. The parameters chosen in this case are varied such that they are expected to result in an increase of the GUE. This is the case if they have one of the following effects:

- Increasing the internal heat exchange (decrease exchanger ΔT)
- Increasing the strong solution concentration (increase P_{evap})
- Decreasing the weak solution concentration (decrease P_{cond} , increase T_{gen})
- Increase ammonia-water energetic separation efficiency

The results of the sensitivity analysis are displayed in Table 3.

Table 3. Results of the sensitivity analysis.

	GUE [-]	% increase
<u>Base case</u>	<u>1.311</u>	<u>0</u>
UA _e (evaporator) increased from 2.73 to 3 kW/K UA _c (condenser) increased from 2.76 to 3 kW/K	1.315	0.36
T _{gen} increased from 150 to 160 °C	1.334	1.75
y _j = increased from 0.96 to 0.98 kg NH ₃ /kg	1.322	0.88
ΔT _{c-f} (pre-absorber) decreased from 10 to 5 K	1.326	1.17
ΔT _{1a-3} (internal HEX CE) decreased from 20 to 10 K	1.317	0.50

It can be seen from Table 3 that the expected effects indeed occurred: the GUE increased when the parameters were changed. It can therefore be concluded that for these parameters the model behaves as expected.

Some notes on Table 3:

- Increasing the heat exchanger effectiveness results in an increase of the GUE. This is true both for the external heat exchangers (evaporator and condenser) and for the internal heat exchangers (Internal HEX CE and Pre-absorber).
- Increasing the temperature of the generator by just 10 degrees centigrade has a relatively large positive effect on the cycle efficiency.
- The effect of better ammonia-water separation in the generator (increasing y_j) on the total cycle efficiency is not very substantial. It furthermore comes at a high capital costs and often equipment weight and might therefore not be worth it.



4.2.2 Trend analysis

As noted in section 4.1 the model holds a lot of information on what is going on inside the heat pump. In order to make some of this information visible, plots of various system variables against outdoor temperatures are shown in this subsection.

Figure 14 shows the variation of the two pressure levels inside the system with outdoor temperature. It can be clearly seen that the evaporation pressure varies across a much wider range than the condensation pressure. This is due to the fact that the building inlet and outlet temperatures are fixed, meaning that the condensation pressure only varies because the water mass flow rates vary somewhat. The evaporating pressure on the other hand has to cope with widely varying outdoor temperatures and will therefore vary much more.

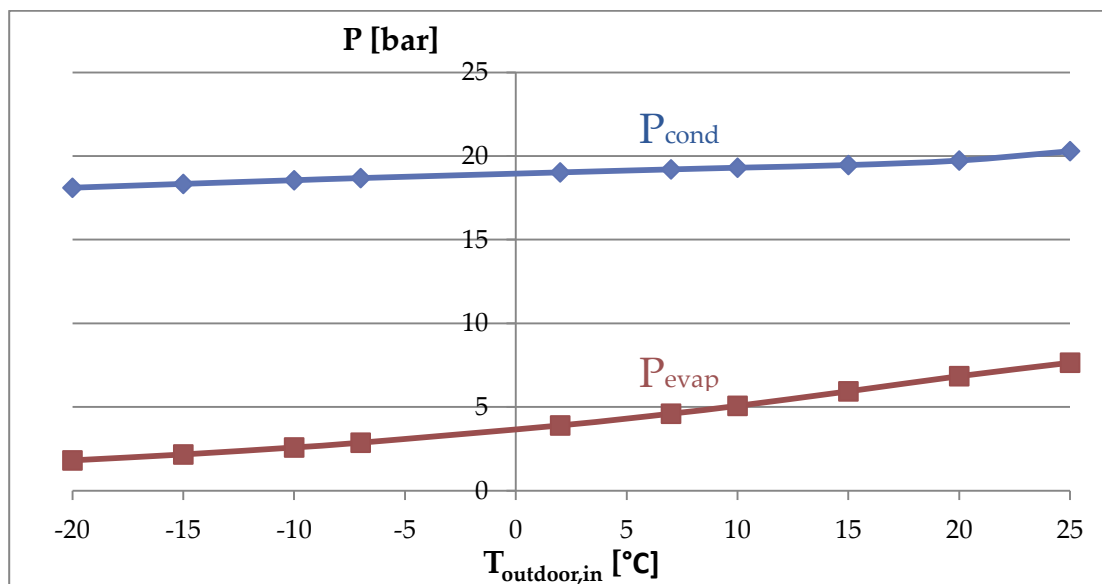


Figure 14. System pressures against outdoor temperature for the GAHP-AR model using the parameter values from Table 16.

In Figure 15 the effect of Figure 14 can be seen once more. In the system the saturation temperature of the strong solution is fixed, just like the generator temperature (saturation temperature for the weak solution). Therefore, the only reason why the concentrations inside the systems vary is because of the variation in the condensation and evaporating pressure. From Figure 15 it is therefore apparent that the strong solution concentration varies with the evaporating pressure, while the weak solution concentration is dependent on the condensation pressure.

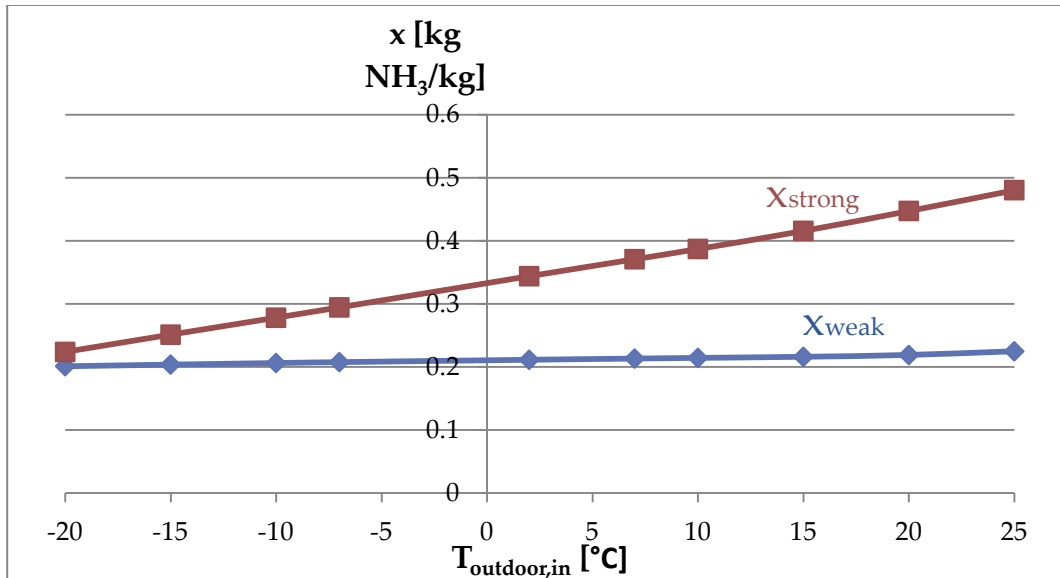


Figure 15. Ammonia concentrations against outdoor temperature for the GAHP-AR model using the parameter values from Table 16.

Another interesting variable to look at is the flow ratio (FR). It is defined as:

$$FR = \frac{\dot{m}_{strong}}{\dot{m}_R}$$

The flow ratio is a measure of how far the concentrations of the weak and strong solution are apart. If these are very close, the flow ratio is large, if they are far apart the flow ratio is small. A large flow ratio means that a large amount of strong solution has to be fed to the generator in order to produce a small amount of refrigerant. The course of the flow ratio with outdoor temperature is shown in Figure 16.

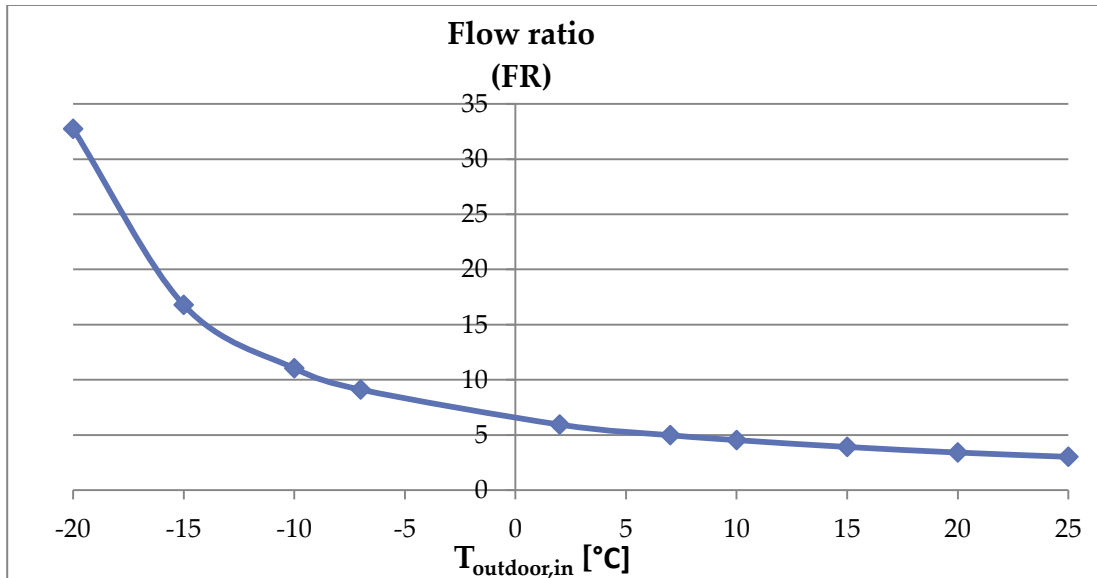


Figure 16. Flow ratio against outdoor temperature for the GAHP-AR model using the parameter values from Table 16.

The next interesting quantity to look at is the amount of heat exchanged, both externally and internally. For this reason the plots of Figure 17 are useful. In this plot the amounts of heat exchanged with the building heating water (blue lines), the amount of heat extracted from the outside (green line) and the amount of heat exchanged internally (purple, red and orange lines) are shown. The generator heat duty is constant, since the burner is unable to modulate and its efficiency is assumed constant. From these plots the effect of the flow ratio is apparent. At low temperatures, when the flow ratio is high, the refrigerant flow is low. This means that the condenser and the evaporator do not exchange a lot of heat with building heating water and outdoor air respectively. Therefore, the absorber is mainly responsible for heating up the building heating water. As the outdoor temperatures rise, the flow ratio decreases. In Figure 17 this is visible as a rise in the condenser and evaporator duties. With rising temperatures the condenser takes an increasingly larger part of the heat duty.

It can be seen from the lines of the pre-absorber, rectifier and internal heat exchanger that the internal heat duties will also rise as the heat pump's heating power increases. Due to the scale of the chart it is not visible that the amount of heat exchanged in the internal heat exchanger almost triples when going from an outdoor temperature of -20°C to 25°C . This is due to the strong increase in refrigerant flow rate from 2.2 g/s to 15.8 g/s respectively. The same effect is visible in the rectifier, where the strong solution has to remove more and more heat to condense the vapour coming from the generator in order to produce the refrigerant needed.

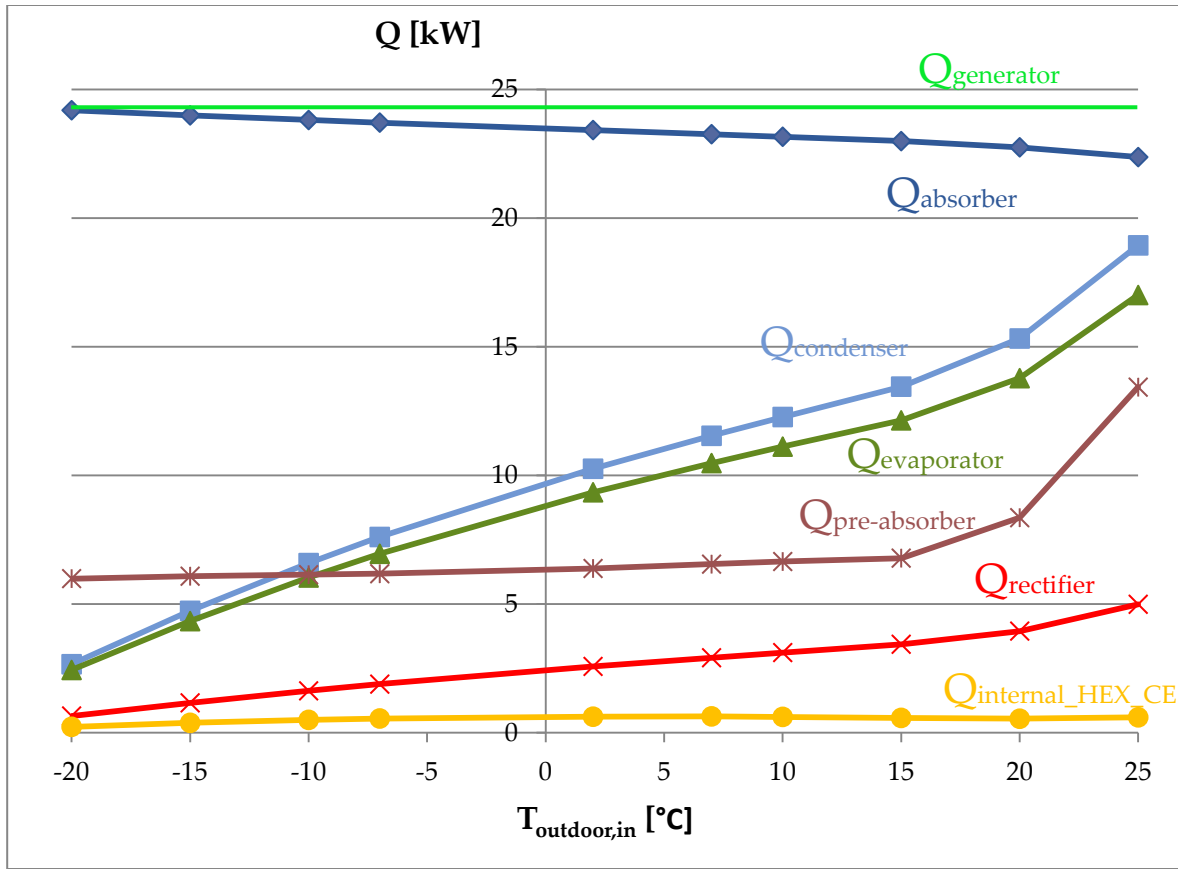


Figure 17. Various internal heat duties against outdoor temperature for the GAHP-AR model using the parameter values from Table 16.

4.3 Quantitative validation - Comparison to manufacturer data

It is important to have a look at the match between model and manufacturer data. To this end the model was run for the same set of outside temperatures it should be able to cope with according to its manufacturer. Its heating power was compared to the heating power quoted by the manufacturer. All parameters are kept at their standard values during the simulation, which are given in Table 2 of section 4.1. The results of the quantitative validation are shown graphically in Figure 18. It can be seen that there are some deviations. The largest deviation of 9.8 % occurs at $T_{\text{outdoor,in}} = -10^{\circ}\text{C}$ and the smallest deviation is 3.2 % for $T_{\text{outdoor,in}} = 2^{\circ}\text{C}$.

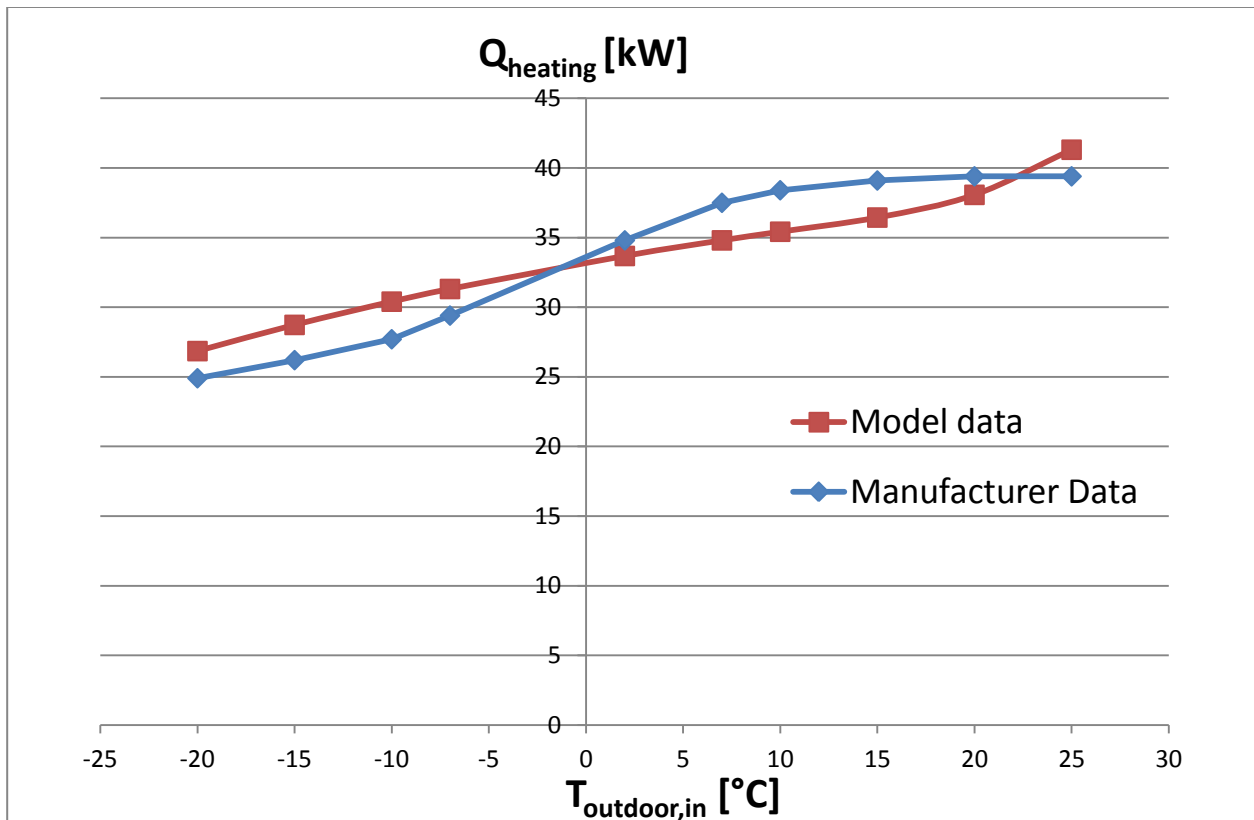


Figure 18. Comparison of the model results to manufacturer data for heating the building heating water from 35 °C to 45 °C.

5 Modelling approach – heating system model

The model of the GAHP-AR heat pump described and validated in chapters 2 and 3 offers a large amount of detail on what is going on inside the heat pump. In reality a heat pump is never the only component in a building heating system. Therefore a heating system model was built. It is explained in this chapter. In the first section the model input is described. The second section is on the details of the heating system model. In the third section the characteristics of the various heat sources are described.

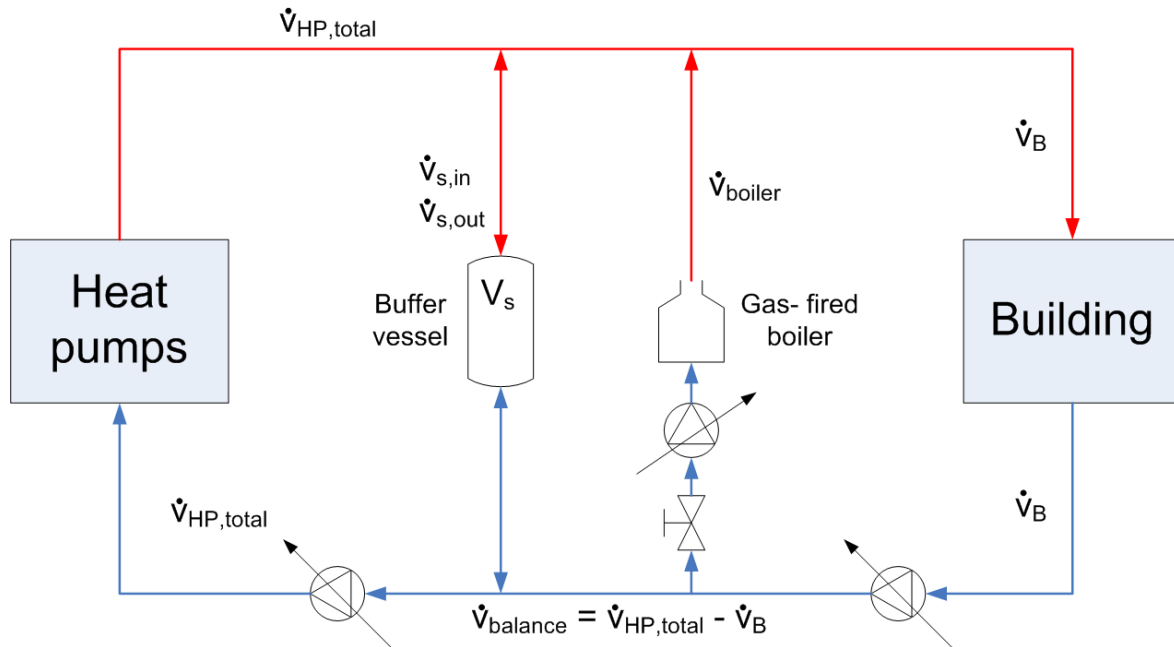


Figure 19. Building heating system used for the heating system model. The 'v dots' in the figure indicate volumetric flows, while V_s indicates the volume of the buffer vessel.

Figure 19 shows the system layout used for the model under consideration. The lines in the figure indicate water flows, where a blue line means cold water and a red line heated water. Besides the heat pump and the building the system consists of a set of gas-fired boilers, some variable-flow pumps and a buffer vessel. The heat pump sub-system consists of a set of gas absorption heat pumps which can only be on or off: they cannot function under part-load. Therefore the amount of heat they produce will never exactly match the amount of heat required by the building. For this reason the system has a buffer vessel, which stores the surplus heat produced. This heat can be used in case the heat pumps produce too little heat. If the heat pumps and the buffer vessel together cannot match the heat demand of the building, the balance will be produced by gas-fired boilers.

It should be noted that the layout drawn in Figure 19 utilizes a control system which uses flow measurements at various locations in order to meet the demands of the building at any time. Since the water entering the gas boiler system is at the lowest temperature level encountered in the

system, the gas boiler is able to function at its highest possible efficiency level. For economic reasons, most systems in practice have a slightly different layout from Figure 19. The gas boilers are then placed behind the heat pumps where they raise the temperature of the water coming from the heat pumps to the desired level if necessary. This system layout enables the use of temperature sensors instead of flow sensors, which is beneficial economically because these are a lot cheaper.

5.1 Model input

The model input consists of the characteristics of both heat source and the heat sink. The heat source is either air, soil or water from an underground aquifer. The heat sink in this case is the building, where the heat from the heating system is used to provide a comfortable indoor climate.

5.1.1 Heat source characteristics

Air source

In case of an air source heat pump, the outdoor air is providing heat to the building through the heat pump. A graphical representation of an air-source gas absorption heat pump is shown in Figure 20.



Figure 20. Air source gas absorption heat pump used for hot water and space heating.³

The outdoor air temperature and relative humidity vary throughout the year, with the lowest temperatures occurring when the building heat demand is at its maximum value.

³ Picture taken from the Robur website on 30-01-2012:
<http://www.robur.com/products/e3-systems/e3-a/description.html>

Ground source

An alternative to the air-source heat pump is the ground source heat pump. A ground source heat pump uses a horizontal or vertical tube bundle which is buried into the ground. A glycol-water mixture circulates through the tubes to extract heat from the ground and carry it to the evaporator of the heat pump. An artist impression of a ground source heat pump with vertical heat exchanger loops is given in Figure 21.



Figure 21. Ground source heat exchanger loops coupled to a heat pump for space heating.⁴

The advantage of this system over the air-source heat pump is that its heat-source is of a higher temperature throughout the year. This results in a higher evaporator pressure and therefore an increase of the heat pump efficiency. Another advantage of this system is that no defrosting is required, which further increases the seasonal efficiency relative to an air source heat pump. It is also important to note that the cold season the ground can cool down substantially as heat is extracted from it. For this reason the ground heat exchanger can be used to provide cooling for the building in summer.

Water source

A third source of heat can be found in underground aquifers. This system involves a separate hot and cold water reservoir at large depth. In winter the water from the hot reservoir is pumped to the surface to serve as a heat source to the evaporator of the heat pump, cooling down in the heat

⁴ Picture taken from from the website of VEO Energy Systems on 30-01-2012:
<http://vanettenoil.com/id77.html>

transfer process. It is then injected back into the ground to serve as a cold reservoir in summer. A picture of such a water source system is given in Figure 22.

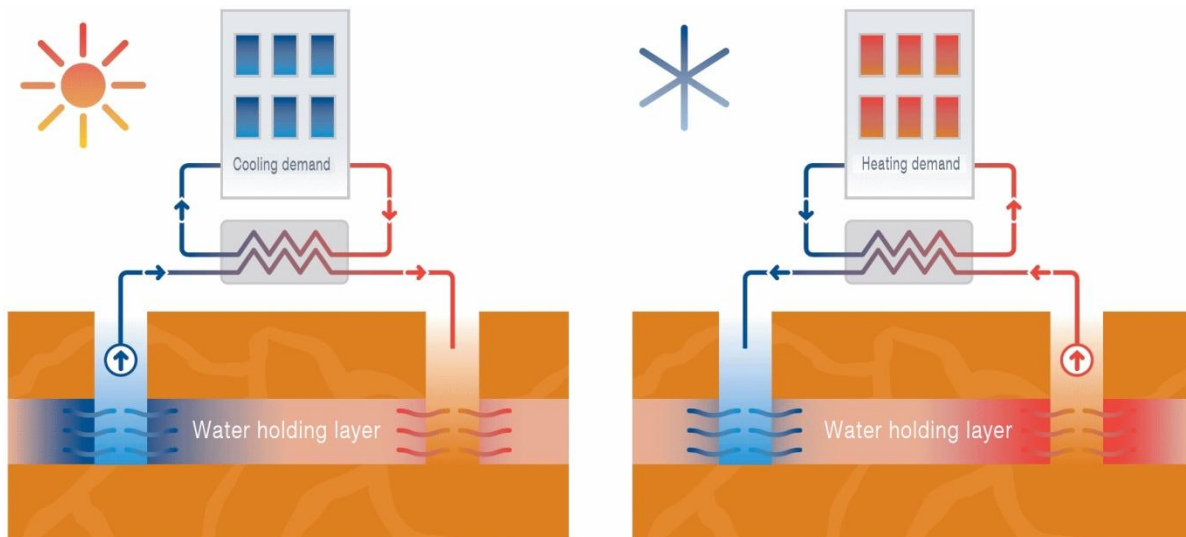


Figure 22. Schematic drawing of a water source heating system, using heat and cold storage in underground aquifers. The summer situation is displayed on the left, the winter situation on the right.⁵

The main advantage of this system over the ground source system is that much less heat is lost through diffusion inside the ground. Both the hot and the cold source are relatively concentrated and therefore there is less surface area for heat loss. This means that the heat source to the evaporator in the cold season will be of a higher temperature, which will positively affect the heat pump efficiency. Provided heat source and cold source are perfectly separated, the cold source will also keep its temperature much better which means less pumping power is required to cool the building in summer. Downsides of the water source system include high initial costs and the requirement to meet regulations on the thermal balance inside the ground.

The model of the water heat source is very simple: a constant temperature of the water coming from the hot source is assumed year-round.

5.1.2 Building heat demand profile

In order to test whether the gas absorption heat pumps are suitable for medium-sized buildings that need much more heating than cooling, the heat demand profile of a nursing home was derived. A very simple, yet often realistic linear relationship between building heating demand and outdoor temperature was assumed (see Figure 23).

⁵ Picture taken from the website of Agentschap NL on 30-01-2012:
<http://agentschapnl-acc.b3p.nl:8080/html/WKO.html>

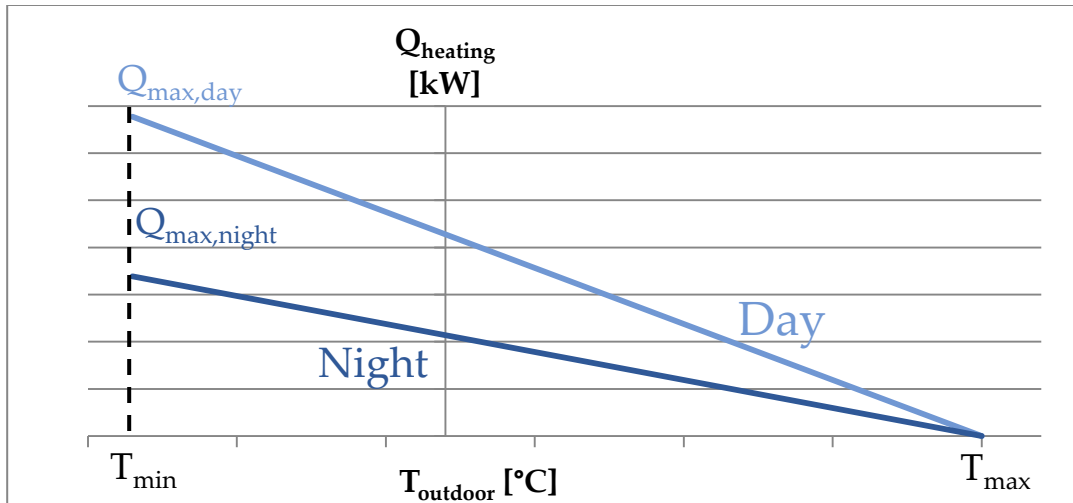


Figure 23. Linear relation between building heating demand and outdoor temperature for a 3000 m² nursing home.

From Figure 23 it can be seen that the heat demand during night time is half the heating demand during day time. This can easily be adopted in the Excel sheet if necessary. Furthermore the graph runs from the minimum temperature -10.5 °C of the climate year under consideration towards a maximum temperature of 18 °C at which the heating system is switched off. It should be noted that Figure 23 is valid for space heating only. Heating needed for the production of hot water used in showers and hot water taps is not taken into account. The characteristics of the nursing home are given in Table 4.

Table 4. Characteristics of the nursing home considered in this report.

Living area [m ²]	3,000
Q _{tot,yearly} [kWh]	639,833
Day from 8-22 hr	

Using the estimation that for our 3000 m² nursing home the yearly heating requirement is 639,833 kWh (SenterNovem, 2007) and combining this with the reference climate year, the heat demand profile for one year can be derived (Figure 24) for $\dot{Q}_{\max,heating,day} = 339$ kW. Details on the calculations involved are given in Appendix J.

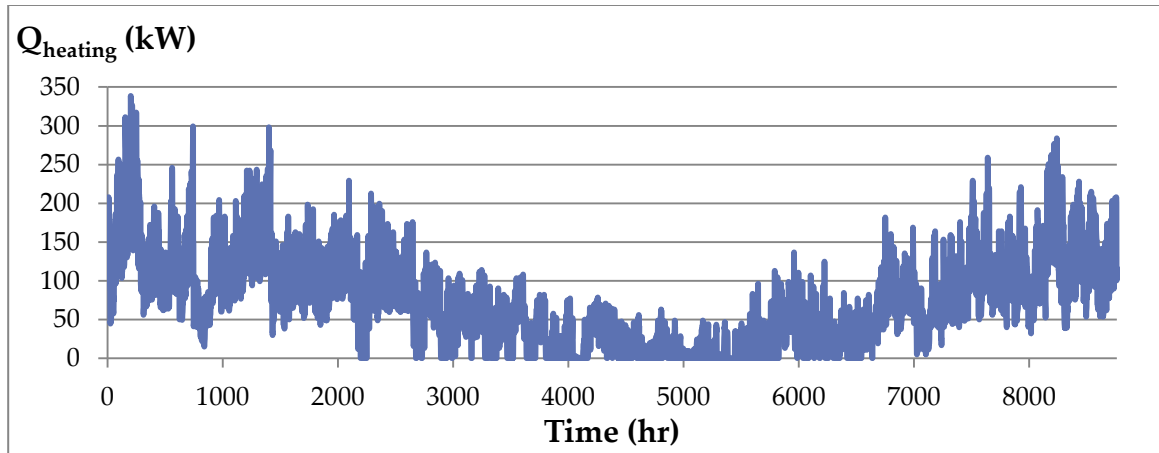


Figure 24. Heat demand profile for the nursing home of Table 4 calculated using the reference climate year RA2008EN.

The number of heat pumps required to meet 50% of the peak heating power is calculated using the heat pump nominal heating power ($\dot{Q}_{HP,nom}$) as obtained from data of manufacturer Robur. The result of this division was rounded to the nearest integer (Table 5).

Table 5. Calculation of the number of heat pumps required to meet 50 % of the building peak heating demand.

	GAHP-AR	GAHP-GS	GAHP-WS
$\dot{Q}_{HP,nom}$ [kW]	35.3	37.7	41.6
N_{HP} [#]	4.80	4.49	4.07
$N_{HP-Rounded}$ [#]	5	4	4
% of $\dot{Q}_{max,day}$ covered	52	45	49

5.2 System model

The system layout was shown earlier in Figure 19. In the present section the heating system is translated into modelling equations. The first sub section deals with the heat pump models, which are a lot more simplified than the model discussed in chapters 2 and 3. In the next sub section the model of the gas boiler is explained. The last sub section is on the model of the control system, which dictates where the heat for the building comes from at any given time.

5.2.1 Heat pump models

In this sub section the models used for the heat pumps are explained. Three different models were built, depending on the heat source used as discussed in subsection 5.1.1. All three models are based on manufacturer data from Robur gas absorption heat pumps.

Reversible air source heat pump (GAHP-AR)

The manufacturer provides data on the performance of its heat pumps under steady state conditions. Fits of these data are used for estimating steady state performance.

Heating power under steady state conditions

Data on the heating power of the heat pump under consideration were obtained from heat pump supplier Techneco. Fits of heat pump heating power with outdoor temperature at the four different supply temperatures were obtained from these data. All fits are third order polynomials and all R^2 values exceed 0.99. The fits are given in Appendix I.

Because manufacturers commonly quote very optimistic efficiency values for their heating systems, all fitted data are multiplied by the value 0.95 to correct for this.

Cycling losses

Depending on the building heat demand, the control system will switch one or more heat pumps on or off on a regular basis. It is known that Robur absorption heat pumps have start up times ranging from 15 -20 minutes (Beekman, 2012). This means there are energy losses associated with this start-up. These energy losses are calculated and taken into account in the model by multiplying a fixed energy loss per start-up with the number of starts. These losses for the various heat pump types are calculated in Appendix E

Defrosting losses

At low temperatures frosting of the heat pumps evaporator becomes a serious issue. Therefore, regular defrosting is necessary at temperatures lower than 6°C and relative humidity values higher than 50% (Luigi, 2000).

Extensive research has been done on the defrosting of vapour-compression heat pumps. Unfortunately, this is not true for absorption heat pumps. The effect of defrosting on heat pump efficiency therefore had to be estimated by combining data from various sources. The following facts about defrosting were used for the estimations:

- During defrosting the heat pump is able to deliver about 50% of its nominal power (Techneco, 2011a).
- Defrosting should last 15 – 20 minutes at a maximum (Beekman, 2012).
- The defrosting frequency at 0°C and a temperature difference of 10 K across the evaporator is about 3-4 hours (Infante Ferreira, 2012).
- The relation between outdoor temperature, relative humidity and frost layer thickness is known for evaporators of vapour-compression heat pumps (Guo et al., 2008).

Using these facts, the following formula for the heat pump heating power at temperatures below 5 °C and relative humidity values above 65% was derived:



$$\dot{Q}_{\text{heating}} = \underbrace{\dot{Q}_{\text{heating,normal}} \cdot (1 - DCF \cdot f_{\text{def}} \cdot t_{\text{def}})}_{\text{during normal operation}} + \underbrace{0.50 \cdot \dot{Q}_{\text{heating,normal}} \cdot DCF \cdot f_{\text{def}} \cdot t_{\text{def}}}_{\text{during defrosting}} \quad (5.2.1)$$

The defrost frequency factor f_{def} is taken to be 0.25 hr^{-1} (1 defrosting cycle every 4 hours for the reference conditions of 0°C and 75% relative humidity). For the defrosting time (t_{def}) 0.25 hrs is taken (15 minutes). The defrost correction factor (DCF) is calculated from the data in Figure 10 of the article on defrosting (Guo et al., 2008) which is reproduced in Figure 25 below. The reason for the peak at 0°C for all curves is that there are two non-linear competing phenomena that affect frost formation. The first is the absolute humidity, which rises when the air temperature is increased at fixed relative humidity. This will enhance frost formation. The other effect is the increase in evaporating temperature when the outdoor air temperature rises. This will result in less frost formation. The defrosting factor has values around unity. It makes sure the defrosting frequency is increased or decreased relative to the reference conditions of 0°C and 75% relative humidity.

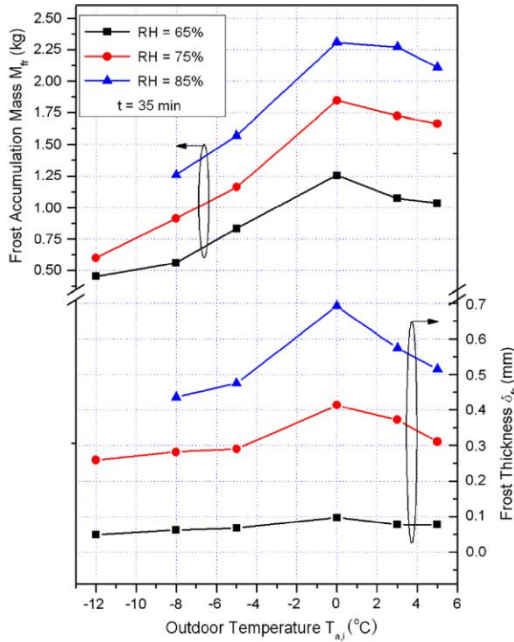


Figure 25. Reproduction of Figure 10 from the article of Guo, Chen et al. 2008. The figure shows the frost accumulation thickness and mass at the evaporator of an air source vapour-compression heat pump after 35 minutes operating time. The upper three curves are used for the calculation of the defrost correction factors (DCF's).

The data points from Figure 25 are extrapolated linearly towards 100 % relative humidity. The corresponding set of data points consists of data on frost accumulation mass with outdoor temperature with four different outdoor relative humidity values. These data points are normalized relative to the point at 75% relative humidity and 0°C . The results after normalization are given in Table 6.

Table 6. Defrost correction factors (DCF's) used in the defrost correction calculations.

Temperature [°C]	RH [%]			
	65	75	85	100
-12	0.26	0.34	0.41	0.53
-8	0.32	0.50	0.69	0.96
-5	0.46	0.64	0.86	1.14
0	0.69	1.00	1.25	1.67
3	0.59	0.93	1.22	1.71
5	0.57	0.90	1.14	1.58

As can be seen in Table 6 there are no values for relative humidity values lower than 65%. Because there is no data on humidity values lower than 65%, it is assumed that no defrosting is needed below 65%, which is a simplification of reality. For humidity values between 65% and 100% and temperatures between -12 and 5 °C, linear interpolation in Matlab was performed with use of the so-called *table lookup* function. This function makes it possible to calculate defrost correction factors for values in between the tabulated values.

Ground source (Robur GAHP-GS-LT) and water source (Robur GAHP-WS)

The calculation methods for the energy use of the ground source and water source heat pumps are very comparable to those used for the air source heat pump. Manufacturer data relating heating power to building supply temperature and ground source or water temperature were fitted. The results are shown in Appendix I.

NOTES ON MANUFACTURER FITS

- 1) It should be noted that the same correction factor of 0.95 is applied to these fits as was done for the GAHP-AR before using them in the model.
- 2) Because the evaporator of the ground source heat pump does not use air to extract the heat from, no frosting of the evaporator occurs. Therefore no defrosting losses occur.
- 3) The cycling losses for the ground source and water source heat pumps are calculated completely analogously to those for the air source heat pump. Information on these losses is given in Appendix E.

Calculation of the amount of cold stored

Ground source and water source heat pumps use geothermal heat to evaporate their refrigerant. The water-glycol mixture (ground source) or water from the aquifer (water source) cools down in this process before it returns to the depths of the earth. In this way cold is 'stored' in the earth for the heating season. In order to find out how much cold is stored, the evaporator duty has to be calculated during the simulation and integrated with respect to time. This can be done easily since

an energy balance on the entire absorption heat pump learns that only a few components of the heat pump exchange heat with the surroundings. These include:

- 1) Generator: heat is added to the system by burning natural gas
- 2) Evaporator: heat is extracted from the ground or from an underground aquifer
- 3) Absorber: heat is given off to the building heating water
- 4) Condenser: the water coming from the absorber is heated up further

The energy balance thus becomes:

$$\dot{Q}_{generator} + \dot{Q}_{evaporator} = \underbrace{\dot{Q}_{absorber} + \dot{Q}_{condenser}}_{\dot{Q}_{heating}} \quad (5.2.2)$$

The amount of heat extracted from the heat source (ground or aquifer) can now be calculated easily, since the burner heating power is constant and known:

$$\dot{Q}_{evaporator} = \dot{Q}_{heating} - \dot{Q}_{generator} \quad (5.2.3)$$

The heat added to the system through the generator is calculated using the burner efficiency value from Table 2:

$$\dot{Q}_{generator} = 0.94 \cdot \dot{Q}_{gas,added} \quad (5.2.4)$$

5.2.2 Gas-fired boiler model

Like for a heat pump, the efficiency of a gas-fired boiler also varies with the building return temperature. The relation between the efficiency (based on LHV) and the temperature of the water coming from the building is given in Figure 26.

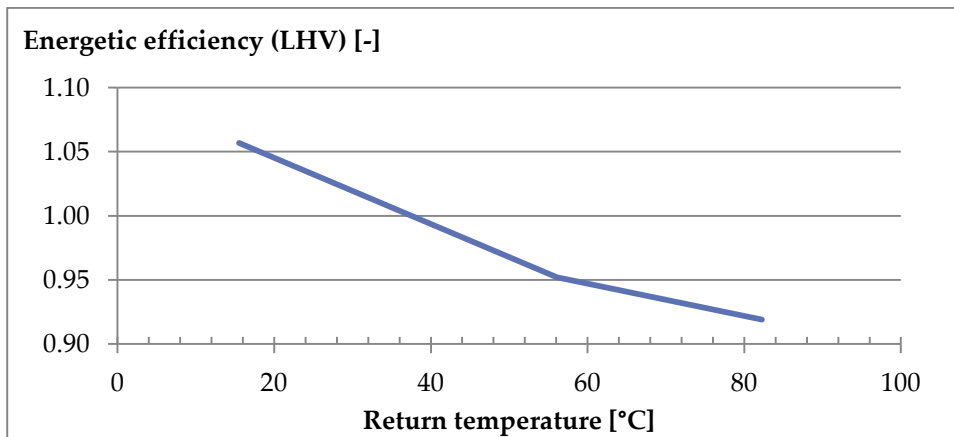


Figure 26. The relationship between building water return temperature and boiler efficiency based on the lower heating value (LHV). Adapted from a presentation held by Jim Cooke (Cooke, 2005).

The graph of Figure 26 was fitted, giving the equations of Appendix K as the result.

5.2.3 Control system model

In order to make sure there is never a surplus or a shortage of heat, a control system is in place.

This control system has to make sure that:

- 1) The flow of hot water into the buffer vessel stops if it is completely filled with hot water
- 2) The flow of hot water out of the buffer vessel stops if the buffer vessel does not contain any more hot water
- 3) The gas-fired boilers are fired up when a shortage of heating power is at hand
- 4) The heat pumps supply as large a percentage of the total heating load as possible
- 5) The heat pumps do not undergo too many on-off cycles in a short period of time

These demands are fulfilled as follows:

- 1) Calculate how many heat pumps are needed to fulfil the heating demand
- 2) Calculate the difference between the amount of heat supplied and the amount of heat delivered by the heat pumps.
 - If the heat pumps supply more heat than needed by the building, the surplus is stored in the buffer vessel, unless there is not enough space left in the buffer vessel in which case one heat pump less is started up.
 - If the heat pumps supply less heat than needed by the building, the control system checks whether the buffer vessel still has enough hot water left to fulfil the balance between supply by the heat pumps and demand by the building. If this is the case, water from the buffer vessel is supplied for as long as possible. If this is not the case, the boilers will supply the balance.
- 3) The control system further checks how much time has elapsed since a heat pump was stopped for the last time. If a heat pump should be started up to meet the heat demand, but it has been closed down less than two hours ago, it will not be started up and the buffer vessel or the gas-fired boiler will make sure there is a balance between supply and demand.

The whole procedure outlined in the paragraphs above can be summarized conveniently in a figure. This graphical representation is shown in Figure 27. This figure is thus a graphical representation of the control system belonging to the heating system of Figure 19 on page 33.

temperature is assumed stable all year long. The ground source temperature is somewhere in between these two extremes: it varies slowly with the seasons, depending on the amount of heat withdrawn from the ground by the heat pump system.

Outdoor air – Robur GAHP-AR heat pump

For the hourly data on outdoor temperature and relative humidity, information from the Koninklijk Nederlands Meteorologisch Instituut (KNMI) is used (KNMI, 2012). The data used are from the climate station in De Bilt, the Netherlands. The climate year used is named RA2008EN. This reference climate year is composed of months from the years 1986 to 2004 according to NEN5060 for building energy use simulations.

Ground – Robur GAHP-GS heat pump

The ground source temperatures depend on the location of the ground heat exchanger and the amount of heat extracted from the ground in winter and the amount of cold required in summer. A representative vertical ground source temperature profile for The Netherlands was obtained from a TNO report (Van Kampen, 2006). This graph was fitted to a fourth-order polynomial as can be seen from Figure 28.

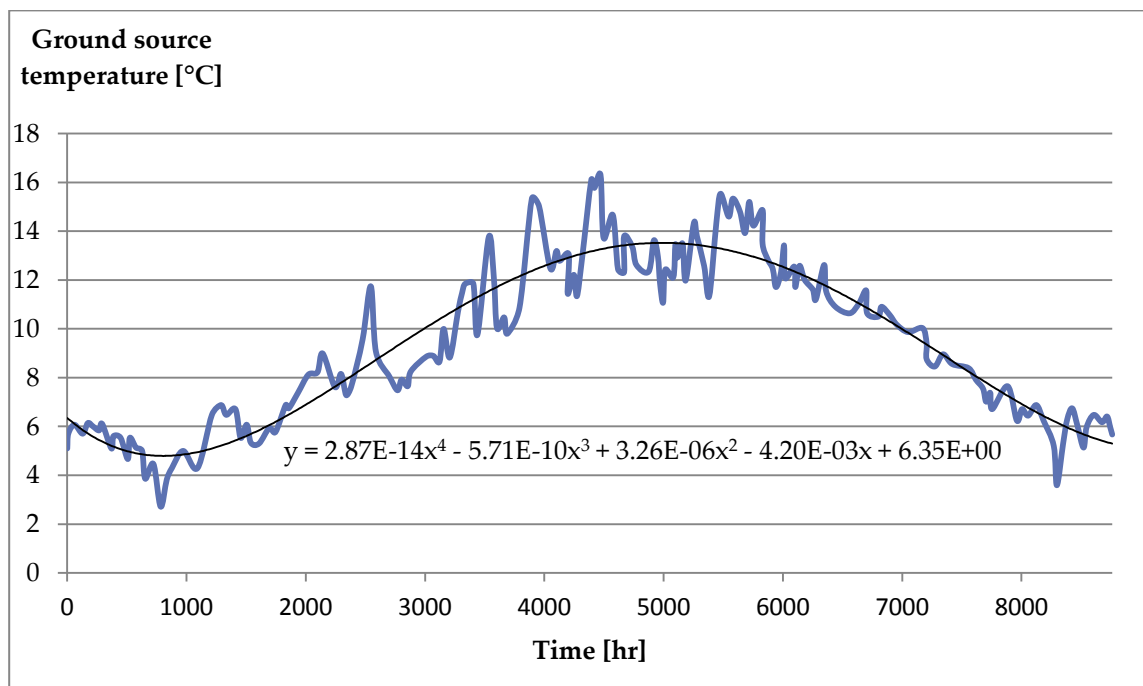


Figure 28. Ground source temperatures representative for a heat pump system using a closed loop vertical heat exchanger. The black line is the trend line fitted to the data. The graph is valid for vertical heat exchangers 35 m in length and for a depth of about 75 m below the surface of the earth.

It can be seen from Figure 28 that the fit only reproduces the global trend of the ground source temperature; the extremes are averaged out. This does not affect the accuracy of the calculations too much however, because the way the ground source temperature varies in reality is different

from system to system. A rough trend is needed in order to be able to estimate the system efficiency.

The water-glycol mixture that is used to extract heat from the ground has to be pumped through the heat exchanger coils below the earth's surface. The pumps required for facilitating this use electrical energy. The power requirements for these pumps were estimated using rules of thumb for the pressure losses inside the coils. The results are expressed as the electrical energy requirements for the pump divided by the amount of thermal energy extracted from the ground:

$$E_{ratio_pumps,GS} = 0.047 \text{ kW}_e / \text{kW}_{th} \quad (5.2.5)$$

Using this ratio the energy required to drive the pumps can be calculated and is taken into account in the model. More details on how the energy ratio given above is calculated can be read in Appendix D.

Underground aquifer – Robur GAHP-WS heat pump

The water source heat pump system makes use of a hot and a cold source far underneath the surface of the earth. If both sources are well separated from each other and thermal diffusion inside the ground is neglected, the temperature of the hot source will remain relatively stable throughout the year in this case. A constant temperature of 15 °C is used for the present calculations. This value was obtained using data from the Royal Haskoning building monitoring system.

The water from the underground aquifer has to be pumped to the surface in order to supply heat to the evaporator. The amount of electrical energy used by the pumps to accomplish this is estimated using the methods outline in Appendix D. The energy ratio of pump work to heat extracted from the ground becomes:

$$E_{ratio_pumps,WS} = 0.022 \text{ kW}_e / \text{kW}_{th} \quad (5.2.6)$$



6 Model validation - Heating system model

In order to check whether the results from the calculations are reasonable, the model for the GAHP-AR air source heat pump is run for several different cases. The models for the GAHP-GS and GAHP-WS were based on the GAHP-AR model. They differ only from the GAHP-AR model in the manufacturer fits used, the fact that the cooling capacity of the ground source is calculated and the absence of the defrosting correction. Therefore, these models will not be validated separately.

Section 6.1 presents a qualitative validation of the results by introducing some deliberate changes and checking the effects on several model outputs. In the second section of this chapter the results are validated quantitatively: by comparison to what is known from the Royal Haskoning building monitoring system.

6.1 Qualitative validation

For all calculations performed here and in Chapter 7 the inputs are as described in section 5.1. The parameters for the reference case are as given in Table 7.

Table 7. Parameter values used for the reference case of the GAHP-AR heat pump heating system model.

$T_{w,building,in}$ [°C]	45
ρ_w [kg/m ³]	992
$c_{p,w}$ [kJ/kg·K]	4.1797
$\eta_{el,grid}$ [-]	0.42
$Q_{gas,added}$ [kW]	25.7
$W_{el,aux,HP}$ [kW]	0.9
t_{def} [hr]	0.25
f_{def} [hr ⁻¹]	0.25
V_s [m ³]	1
N_{HP} [#]	5
Δt [mins]	7.5

6.1.1 Sensitivity analysis

In order to validate the model, three different variations to the reference case are run for an entire year of simulation time:

- 1) No correction for decreased performance during defrosting
- 2) Three times the standard buffer size
- 3) Building inlet temperature of 60 °C (instead of 45 °C)



For these three cases, the following system quantities are plotted:

- 1) Number of heat pump starts
- 2) Percentage of total energy provided by the heat pumps
- 3) Seasonal performance factor of the entire system (= heat delivered divided by primary energy used)
- 4) Seasonal performance factor of the heat pumps
- 5) Seasonal performance factor of the boilers (equals their efficiency based on lower heating value)

The percentage deviations with respect to the reference case are calculated and shown in Figure 30 and Figure 29.

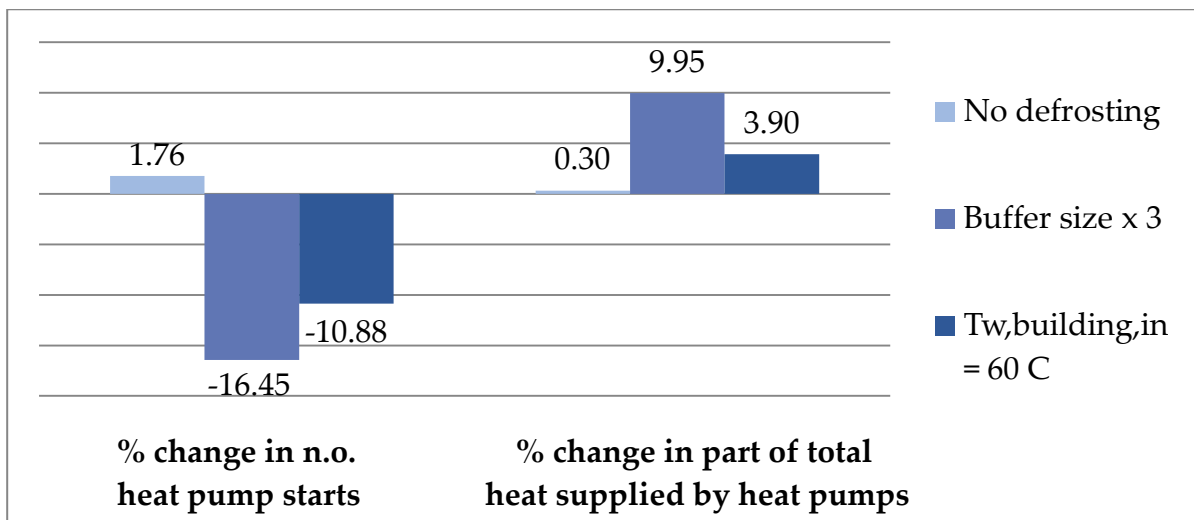


Figure 29. Qualitative validation: effect of several variations to the reference case on heat pump cycling and the share of the heat pumps in the total amount of heat produced. Percentage changes with respect to the reference case are shown.

NOTES ON FIGURE 29:

- 1) No defrosting correction increases the number of heat pump starts and thereby slightly the share of the heat pumps in the energy supplied to the building. The reason behind this could be that the increased heating power of the heat pumps causes the buffer vessel to be filled faster, meaning the heat pumps will shut down faster and therefore run for shorter periods at a time. After the obligatory two hour shut down period they will start-up immediately since the heat demand is large at times when defrosting is normally required. A heat pump run during the defrosting period will thus last shorter when no defrosting correction is carried out. This means there will be more heat pumps starts.
- 2) If the buffer size is three times as large, the amount of heat supplied by the heat pumps goes up drastically, while the amount of heat pump starts decreases. This is because it

takes much longer for the buffer vessel to be filled and to be drained of hot water which means the heat pumps can long for longer stretches at a time without the need to shut them down and start them up again later. Since the buffers are drained less fast, the chance that they are drained before the heat pumps are allowed to start up again (remember the obligatory two-hour stopping period) is much smaller. This has a positive effect on the total amount of heat supplied by the heat pumps.

- 3) Increasing the building supply temperature means that the heat pumps heating power goes down quite significantly. This means that the 'step size' of the heat pump system goes down. Therefore, heat pumps can be started up earlier and shut down later since the buffer vessel is filled less fast with hot water. This positively affects the amount of heat supplied by the heat pumps while at the same time decreasing the number of heat pump starts.

The effect of the three different deviations from the reference case on the seasonal performance factors is shown in Figure 30.

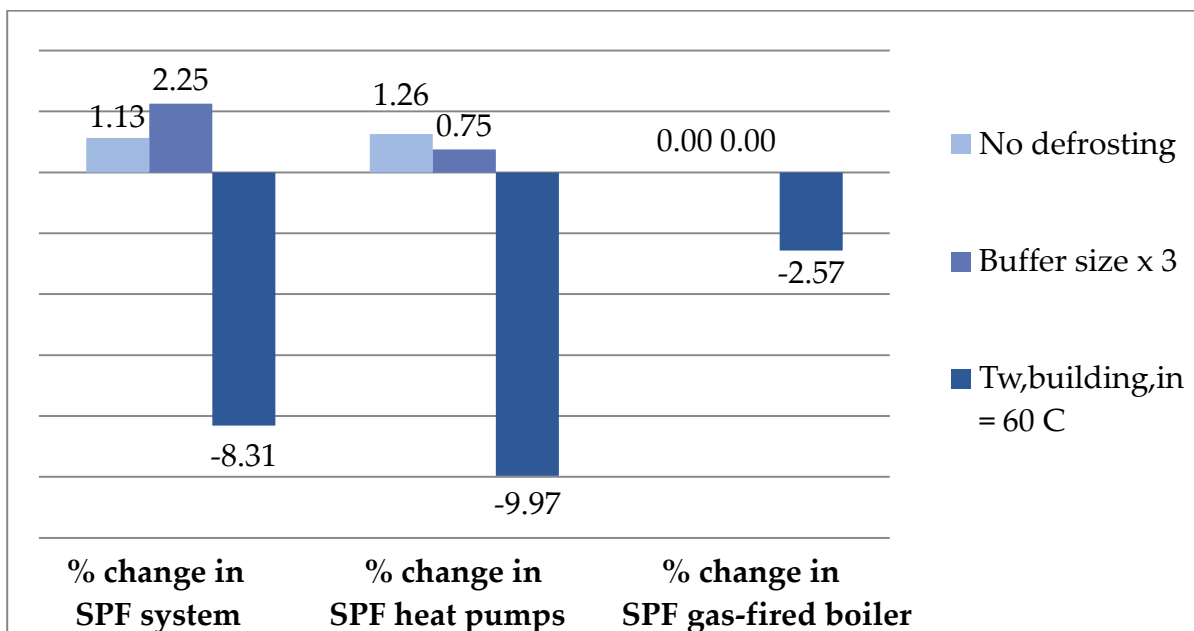


Figure 30. Qualitative validation: effect of several variations to the reference case on the seasonal performance factors. Percentage changes with respect to the reference case are shown.

NOTES ON FIGURE 30:

- 1) The results in Figure 30 show that the defrosting corrections proposed in section 5.2.1 have a relatively moderate effect on the energy efficiency of the system. Averaged over one whole year the SPF of the heat pumps increases by 1.1% if no defrosting correction is carried out. The boiler efficiency is, of course, not affected by the defrosting correction.

- 2) Increasing the buffer size by a factor three has a reasonable effect on the overall efficiency of the system. Its effect on the heat pump efficiency is moderate and probably caused by the decreased number of heat pump starts as was shown in Figure 29.
- 3) Increasing the desired temperature of the water entering the building has a large effect on the performance of the system: it goes down by about 8%, largely caused by the decreased heat pump efficiency. The efficiency of the gas-fired boilers also goes down, because the building return temperature will also go up by increasing its supply temperature.

6.1.2 Influence of simulation step size

A parameter that significantly affects the outcomes of the calculations is the simulation step size. This simulation step size is shown as Δt in Figure 27 on page 44. As explained on that page, the heat pumps will only be started up if the amount of surplus heat they produce is not larger than the amount of heat that can be stored in the buffer vessel when the heat pumps run for a time equal to the simulation step size. In real life the heat pumps will be shut down as soon as the buffer vessel cannot hold any more hot water, which corresponds to an infinitely small step size in the simulation. Therefore, the decreasing the simulation step size will increase the accuracy of the calculations. To show this effect the simulation was run for various step sizes. The results are shown in Figure 31. Note that the horizontal axis shows the number of simulation time steps per hour instead of the simulation step size. A smaller step size thus corresponds to a larger number of time steps per hour.

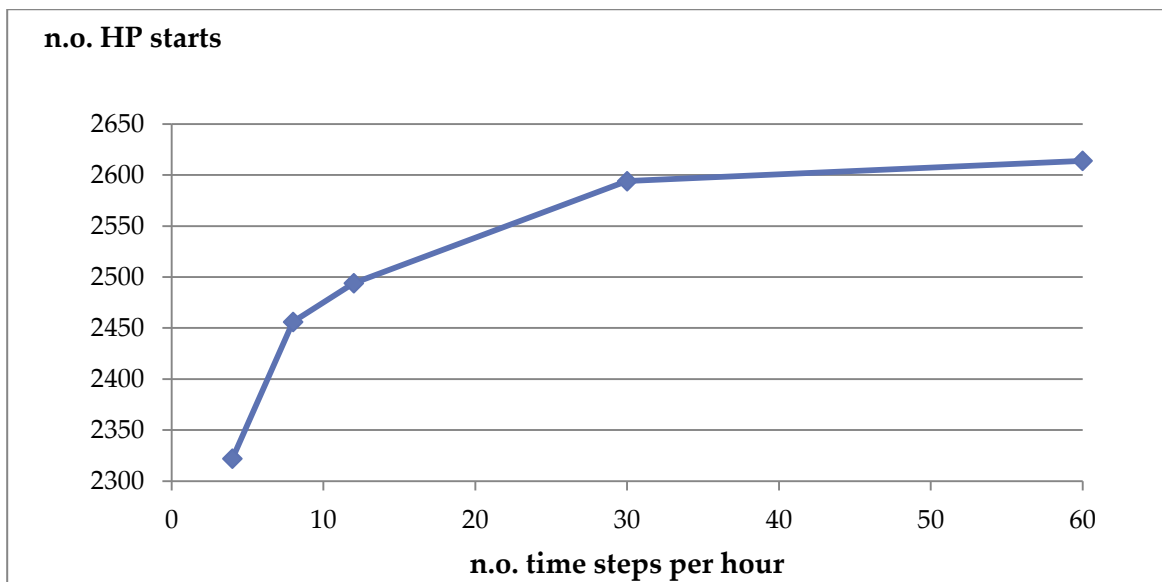


Figure 31. Effect of the size of the simulation time step on the number of heat pump starts for the heating system using the GAHP-AR heat pump. It is important to note that this graph was constructed using a previous version of the model. The absolute values are not accurate therefore. The trend followed will be the same for the newer model versions.

It can be seen in Figure 31 that the larger the number of time steps per hour, the higher the number of heat pump starts becomes. This is expected since more time steps per hour means that the time steps are smaller. There is thus a larger chance that the surplus heat produced by the heat pumps will 'fit in' the buffer vessel during the given time step. The control system will only start-up a heat pump if this is the case. For a smaller time step this situation will occur more often and therefore more heat pump starts will occur.

6.2 Quantitative validation

Since there is not too much known about using gas absorption heat pumps in buildings, there is a lack of quantitative data on this topic. It is known that for vapour compression heat pumps the share in the annual heat production is about 80% when the heat pumps are selected to meet 50% of the peak heating power. The exact value depends on the annual load duration curve. It was shown earlier that in the case of the GAHP-AR heat pump, five heat pumps are required to fulfil about 52% of the peak demand. Table 8 compares the expected to the real share in the total annually delivered heat.

Table 8. Quantitative validation: comparison to experience from the RH Building monitoring system.

	RH Building monitoring	Model
% of peak power delivered by heat pumps	50	52.1
% of annual heating needs met by heat pumps	80	82.5

Although Table 8 provides no conclusive evidence that the model is right, it at least shows that its predictions seem reasonable for the system under consideration.





7 Modelling results – Heating system model

In the previous chapter the model for the air source gas absorption heat pump was validated. In this chapter the results of all three models are presented and compared to find out which one performs best from an energetic point of view and how large the differences between the models are (section 7.1 & section 7.2). The next section provides an answer to the question how the primary energy use of the heat pumps is distributed across various categories. For the ground and water source heat pumps it is furthermore interesting to look at the amount of heat they extract from the ground during the heating season (section 7.4).

7.1 Seasonal performance factor

The seasonal performance factor is an interesting output to look at, since it is the only efficiency measure that provides information on the real average primary energy efficiency.

The input parameters from Table 9 are used to generate the simulation results displayed in this chapter. Data on the heating demand and outdoor temperature conditions are obtained from Chapter 5.

Table 9. Parameter values used for all simulations in this chapter.
 Δt is the simulation time step.

	air source	ground source	water source
$\eta_{\text{burner}} [-]$	not used	0.94	0.94
$W_{\text{el,aux,HP}} [\text{kW}]$	0.90	0.47	0.47
$t_{\text{def}} [\text{hr}]$	0.25	not used	not used
$f_{\text{def}} [\text{hr}^{-1}]$	0.25	not used	not used
$N_{\text{HP}} [\#]$	5	4	4
$\rho_w [\text{kg/m}^3]$		992	
$c_{p,w} [\text{kJ/kg}\cdot\text{K}]$		4.1797	
$\eta_{\text{el,grid}} [-]$		0.42	
$Q_{\text{gas,added}} [\text{kW}]$		25.7	
$V_s [\text{m}^3]$		1	
$\Delta t [\text{mins}]$		7.5	

Figure 32 shows the simulation results for high and low supply temperatures. It is clear from this figure that the seasonal performance factors show the expected trends. All systems using heat pumps have a higher seasonal performance factor than the system using only gas-fired boilers. Furthermore the water source heat pump has a higher seasonal performance factor than the ground source heat pump, which can be explained by the fact that the water source system can utilize higher temperature heat than the ground source system. Another reason why the water



source system performs better energetically is that its pumps use less energy for a given quantity of heat delivered to the evaporator than the pumps of the ground source system.

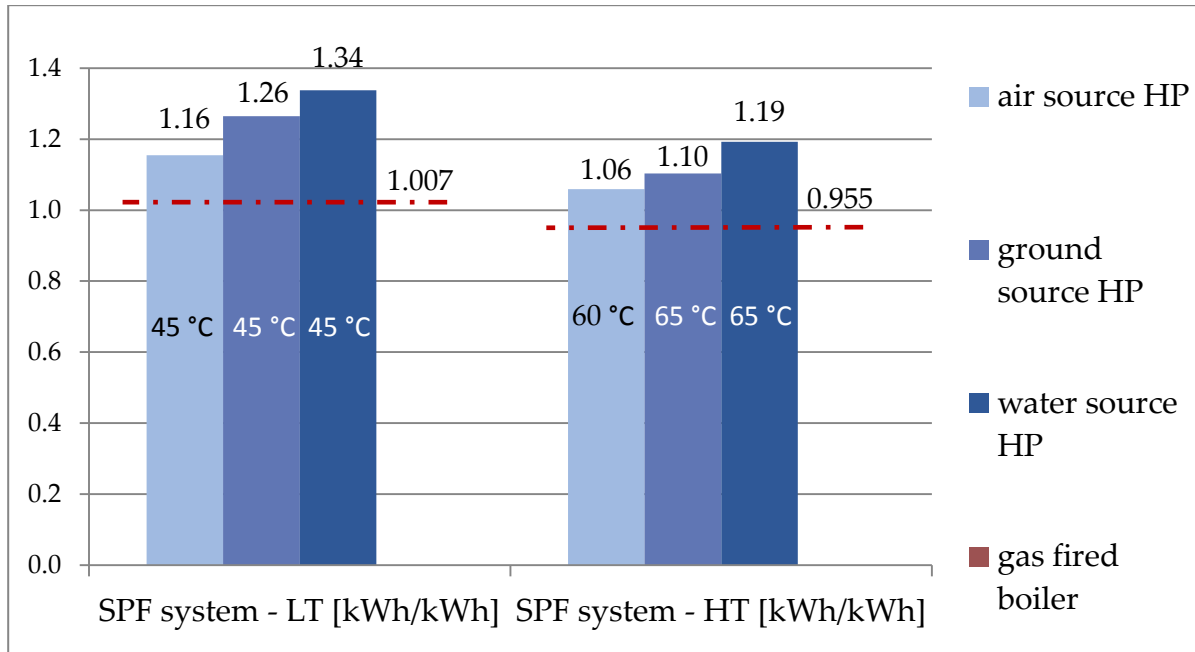


Figure 32. Seasonal performance factors of the air source, ground source and water source heat pump heating systems. The building supply temperature is indicated inside the bars. The SPF of a heating system employing only gas-fired boilers is also shown. All values are based on the LHV. The simulation of the ground source heat pump at high supply temperature was performed using a fit of the Robur GAHP-WS (HT) heat pump.

The high temperatures shown in Figure 32 are the maximum temperatures these heat pumps can deliver. It can be seen that the seasonal performance factors go down as the building supply temperatures go up. It can also be seen that the relative difference between the SPF of the heat pumps and the SPF of the gas boiler goes down; meaning that the decrease of the SPF of the heat pumps is larger than the decrease of the SPF of the gas-fired boiler if supply temperatures go up.

7.2 Total primary energy use – comparison to gas boiler system

It is interesting to compare the total primary energy use of the before mentioned systems to the primary energy use of a gas boiler only system. This is done in Figure 33 for high and for lower supply temperatures.

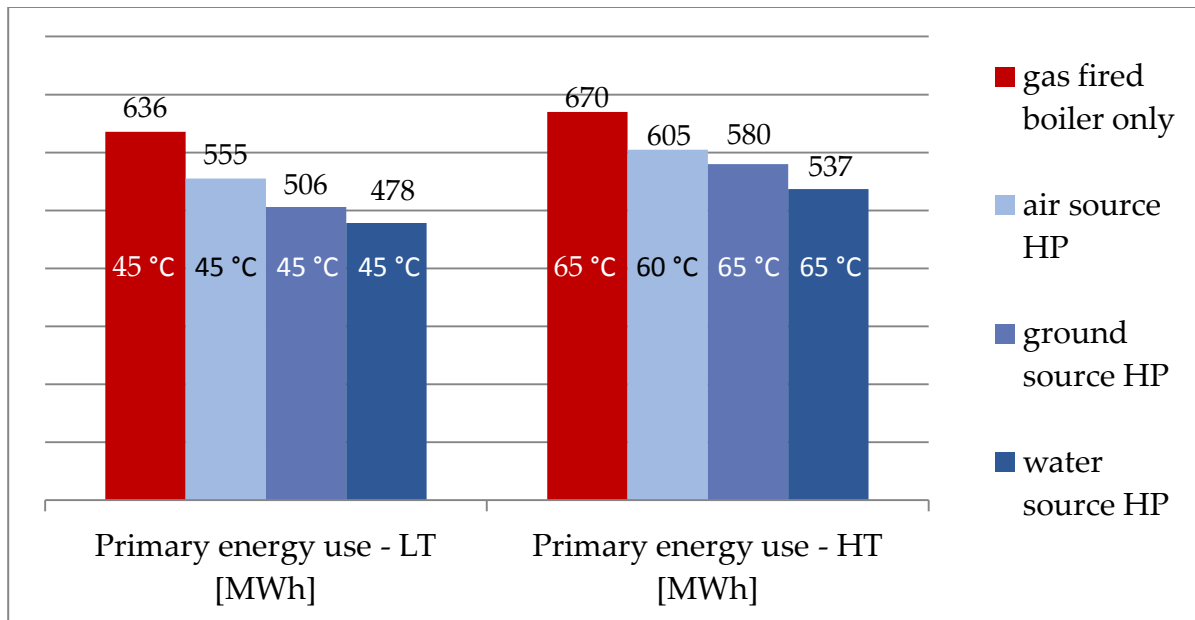


Figure 33. Primary energy use of the air source, ground source and water source heat pump heating systems. The building supply temperature is indicated inside the bars. The primary energy use of a heating system employing only gas-fired boilers is also shown. All values are based on the LHV.

From Figure 33 it becomes apparent that all heating systems using heat pumps use less primary energy than a heating system consisting of only gas boilers. The advantage is not very large in all cases: at low supply temperatures we have 13, 20 and 25 % in case of air source, ground source and water source systems respectively. On the other hand, it is clear that gas absorption heat pumps provide a means to cut down the amount of primary energy used.

When looking at high supply temperatures it becomes clear that the differences between heat pump systems and the gas boiler only system become smaller at elevated supply temperatures. The percentage advantage of the heat pump systems with respect to the gas boiler only system have shrunk to 10, 13 and 20 percent for air source, ground source and water source systems respectively.

7.3 Distribution of primary energy use

As described in section 5.2.1 on the heat pump modelling, the primary energy use of the heat pumps is determined by more than its operational gas use alone. Energy is spent for driving internal auxiliaries like the solution pump, for driving the ground and water source pumps and for starting up the heat pump. The distribution of the heat pumps primary energy use is given in Figure 34.

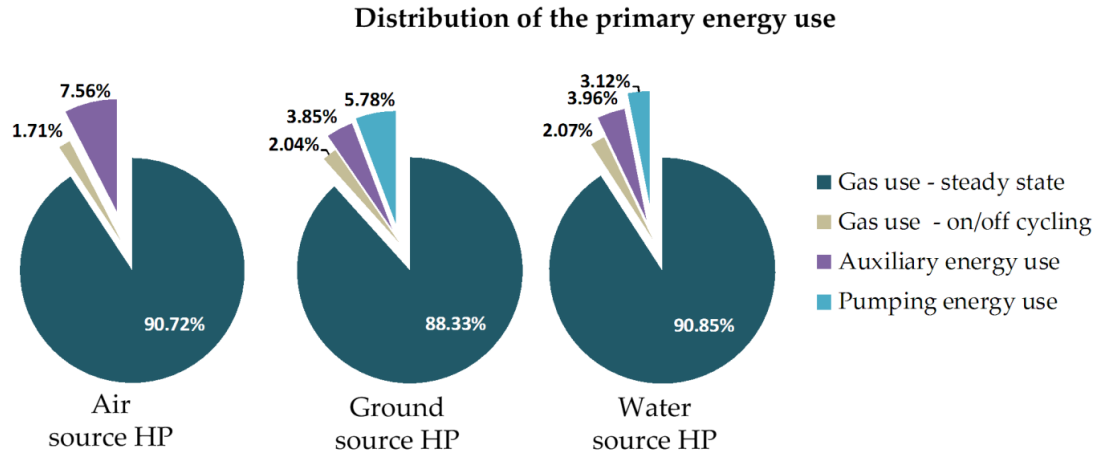


Figure 34. Pie charts showing the distribution of the heat pumps energy use across the various categories for the air source, ground source and water source heat pumps. The building supply temperature is 45 °C for all systems.

It can be seen in Figure 34 that a significant percentage of the heat pumps total energy use is for driving electrical equipment like pumps and fans. The air source heat pump does not require any energy for pumping up water from the ground, since it uses air as the heat source. For the ground and water source systems, driving pumps requires a significant amount of primary energy.

7.4 Cooling capacity produced for summer season

For the ground source and water source heat pumps it is important to know how much heat is extracted from the ground or aquifer during the heating season because this determines the cooling potential in summer. After running the simulations for one whole year with building supply temperatures of 45 and 65 °C the results become as given in Table 10.

Table 10. Cold storage inside the ground or the aquifer reservoir over the course of the whole climate year for two building supply temperatures.

GAHP type	T _{supply} = 45 °C		T _{supply} = 65 °C	
	Cold stored [MWh]	Heat/cold ratio HP	Cold stored [MWh]	Heat/cold ratio HP
Ground source	198.4	2.6	129.6	3.9
Water source	212.7	2.4	163.5	3.2

It is known that at elevated supply temperatures (65 °C), the heating power of the heat pumps will go down as a larger portion of the heat is delivered by the absorber instead of the condenser. This means the evaporator duty will go down as well, resulting in an increase in the heat/cold ratio of the heat pumps. This can be seen in Table 10. Furthermore the ground source heat pump, with its lower efficiency, has a higher heat to cold ratio which is also expected since at lower evaporation temperatures the part of the heat that is extracted at the evaporator and supplied to the condenser decreases, while the absorber duty does hardly change (Figure 17 at page 31).

8 Matching heat pump and application

When energy efficient solutions are desired, there is never 'one size that fits all'. Every application has its own needs and selecting a suitable climate installation requires careful consideration. It is for this reason that this project was initiated and in this chapter the modelling results for the gas absorption heat pumps are put next to the specific demands of a nursing home to check whether the desired match occurs. In the first section the demands for both heating and cooling of the nursing home described in section 5.1.2 are treated. The second section describes and compares different concepts for heating and cooling. Section 8.3 is a discussion of the results from the previous two sections. The last section of this chapter provides some economic considerations on the use of gas absorption heat pumps in practice.

8.1 Demand side

The project aims to find out whether there is a match between the heat-to-cold ratio of nursing homes and gas absorption heat pumps. From internal Royal Haskoning data it is known that the heat-to-cold ratio of a nursing home is about $2.78 \text{ kWh}_{\text{heat}}/\text{kWh}_{\text{cold}}$. From this ratio, the yearly heating and cooling demands for the reference nursing home of Table 4 become (Table 11):

Table 11. Yearly heating and cooling demands for the 3000 m² nursing home of Table 4 on page 37.

Demand side	
Heating [MWh]	640
Cooling [MWh]	230

The value of the cooling demand in Table 11 was calculated by dividing the heating demand from page 37 by the heat-to-cold ratio of 2.78.

8.2 Supply side

8.2.1. Different heating and cooling concepts

In order to find out whether the gas absorption heat pumps have a good match with the heating and cooling needs of nursing homes, they are compared to alternative systems for heating and cooling. The five different concepts considered for the supply side are outlined in Table 12.



Table 12. Different heating/cooling concepts studied. The building supply temperature is 45 °C in all cases. GAHP = Gas Absorption Heat Pump, VCHP = Vapour-Compression Heat Pump.

Supply side		
1	GAHP - air source	Gas-fired boiler Air-cooled chiller
2	GAHP - ground source	Gas-fired boiler Air-cooled chiller
3	GAHP - water source	Gas-fired boiler Air-cooled chiller
4	VCHP - water source	Gas-fired boiler -
5	-	Gas-fired boiler Air-cooled chiller

For the supply side, the systems employing gas absorption heat pumps (systems 1, 2 and 3) are as described in the previous chapter, meaning that the results from the previous chapter are used for these calculations. There is one addition to these systems that has not been discussed before: the air-cooled chiller. The air-cooled chiller is needed because the gas absorption heat pumps using a ground- or water source do not store enough cold underground during the heating season to deliver all the cooling needed in the cooling season, as can be seen by comparing Table 10 on page 56 to Table 11 on page 57. For the air-cooled chiller, a COP of 3.99 was assumed, which is equal to the ESEER of the *Carrier 30RB* 162 kW air-cooled chiller. This is an approximation, since actual part-load conditions during use may differ from the Eurovent conditions for which the chiller was tested (see Appendix F).

The fourth heating concept in Table 12 employs an electrically driven vapour-compression heat pump with a heating COP of 3.5 coupled to heat and cold storage in underground aquifers. The value for the heating COP was obtained from documentation from Climaveneta on the performance figures of the *Prana 0121t* water-source heat pump⁶. The gas-fired boiler was modelled according to Figure 26 on page 42.

Since the water-source vapour-compression heat pump has a heat-to-cold ratio of 1.4 at a COP of 3.5, it will produce too much cold on a yearly basis if it would deliver all heat required by the building. Therefore, it is dimensioned at 36% of the peak heating load. It will then deliver about 50% of the yearly heating energy demand and all of the cooling required. Therefore, this system does not need an additional air-cooled chiller, because all cooling needed in summer can be delivered by direct cooling from the cold storage. This process is called 'discharging the cold source' or 'direct cooling' in the building services jargon.

8.2.2 Results

The different heating and cooling concepts studied differ in how they meet the buildings energy demand. This is shown in Figure 35 for a building supply temperature of 45 °C.

⁶ Condenser in-out 40/45 °C, source water in 10 °C.

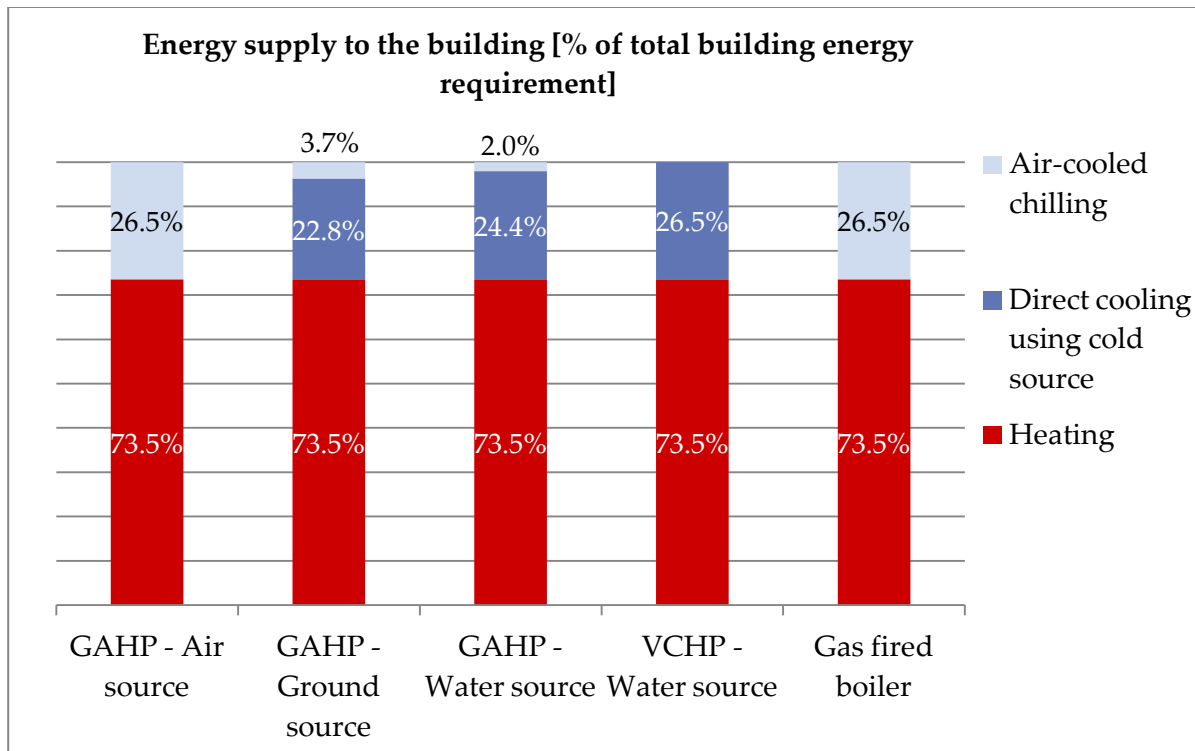


Figure 35. Energy supply to the building using different heating concepts. The building supply temperature is 45 °C.

The total natural gas and electricity consumption of the five concepts of Table 12 are shown in Table 13.

Table 13. Total gas and electrical energy use needed for heating and cooling the nursing home of Table 4 on page 37 for the reference climate year.

	GAHP - Air source	GAHP - Ground source	GAHP - Water source	VCHP - Water source	Gas- fired boiler
Gas use [MWh]	520	467	452	315	636
Electrical energy use [MWh]	72	34	20	102	58

Using the central power generation efficiency from Table 9 on page 53, the results from the simulation summarized in Table 13 can be calculated back to the primary energy requirements. This allows comparison of the different heating and cooling system concepts on an energetic basis. These results are given in Figure 36. From this figure it is apparent that the system using gas absorption heat pumps in combination with thermal storage in underground aquifers (water source) is the most favourable from an energetic point of view. Of all heat pump systems, the air source gas absorption heat pump performs worst. All heat pump systems perform better energetically than the system using only the gas-fired boiler for heating and the air cooled chiller for cooling.

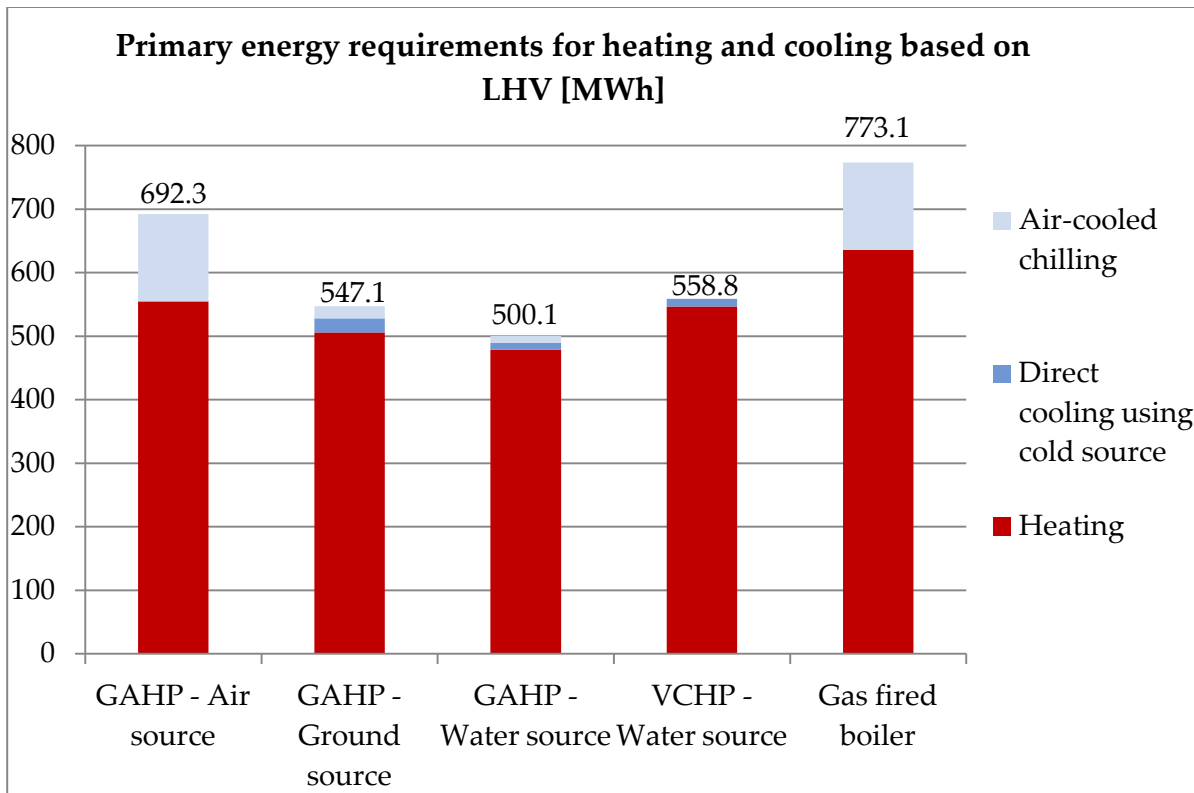


Figure 36. Primary energy requirements for heating and cooling the nursing home of Table 4 on page 37 for the reference climate year. The building supply temperature is 45 °C. The primary energy requirements are based on the lower heating values of the fuels.

8.3 Discussion of energetic results

Looking back at the project goals as defined in Chapter 1 it can therefore be said that the gas absorption heat pump coupled to geothermal storage in underground aquifers provides a good match with the nursing home considered. This can be attributed to the heat pump's high heat-to-cold ratio of 2.4 (at $T_{\text{supply, building}} = 45^{\circ}\text{C}$), which is much higher than the 1.4 encountered in conventional electrical heat pumps (at $\text{COP} = 3.5$) and therefore more favourable.

8.4 Economic considerations

Heat pump systems are not chosen by merely selecting the system with the best energetic performance. Costs also play an important role. This section discusses some generally true aspects of the economics of heat pumps systems using geothermal energy storage: the ground source and the water source heat pump systems.

Investment costs of geothermal energy storage heat pump systems

The total costs of a certain energy conversion system are dependent on three main factors: energy costs, maintenance costs and investment costs. Energy costs can be calculated from the results of section 8.2 and will not be discussed here. Maintenance costs are also not considered in this section.

In the previous section it was shown that the heating and cooling system employing GAHP's combined with geothermal energy storage in underground aquifers is the most energy efficient system for heating and cooling the nursing home considered. It is important to note however, that these water source systems have to comply with Dutch regulations stating that there has to be thermal balance in the ground. In other words: all cold stored during operation in heating mode has to be extracted again from the ground in the cooling season. These regulations do not hold for so-called closed systems (vertical ground loop heat exchangers); thermal imbalance is allowed for ground source systems. However, this situation can only work as long as there are not too many other geothermal energy storage users in the vicinity of the ground loop heat exchangers, because otherwise the ground would cool down with the passing of the years.

Matching the heat-to-cold ratio of the heating system to the building

One way to maintain the thermal balance in the ground is to make sure that the heat pump system does not 'produce too much cold'. Therefore, the heat-to-cold ratio of the system has to match the heat-to-cold ratio of the building. This match can be obtained by adjusting the part of the heating demand delivered by the heat pumps, with the remainder done by gas-fired boilers. Provided the heat-to-cold ratio of the heat pumps is always lower than that of the building, the following formula applies:

$$\text{max \% of total heating energy delivered by HP's} = \frac{\text{heat-to-cold ratio HP's}}{\text{heat-to-cold ratio building}} \times 100\%$$

If this maximum percentage of heating energy delivered by the heat pumps is exceeded, not all cold stored inside the underground cold reservoir by the heat pumps is needed by the building. This means there is no thermal balance in the ground and that is undesirable.

Fixed and variable costs for geothermal storage in aquifers

The costs of geothermal storage in underground aquifers can be subdivided into two groups: fixed costs and variable costs. The fixed costs do not depend on the size of the reservoir used, while the variable costs do vary with reservoir size. Since the fixed costs are generally high, it is only favourable to use aquifer energy storage when the sources are large (~75 kW reservoir capacity). Systems needing reservoir capacities smaller than 25 kW generally use vertical ground loop heat exchangers which do not have large fixed costs, while systems in between these two values can use either of both depending on the exact cost calculations. (Woldring, 2012)

Sizing of the aquifer reservoirs

When it comes to the sizing of the underground reservoirs there are two extremes:

- 1) Reservoir capacity is equal to the maximum evaporator duty of the heat pump system
- 1) Reservoir capacity is equal to the maximum cooling power needed in summer



Both extremes have their own advantages and disadvantages. For option 1 the advantage is that the variable costs for the reservoir are small, but the disadvantage may be that not all cold can be extracted from the cold source in the cooling season, leading to thermal imbalance. Whether or not this occurs depends on the shape of the annual load duration curve for cooling, an example of which is given in Figure 37.

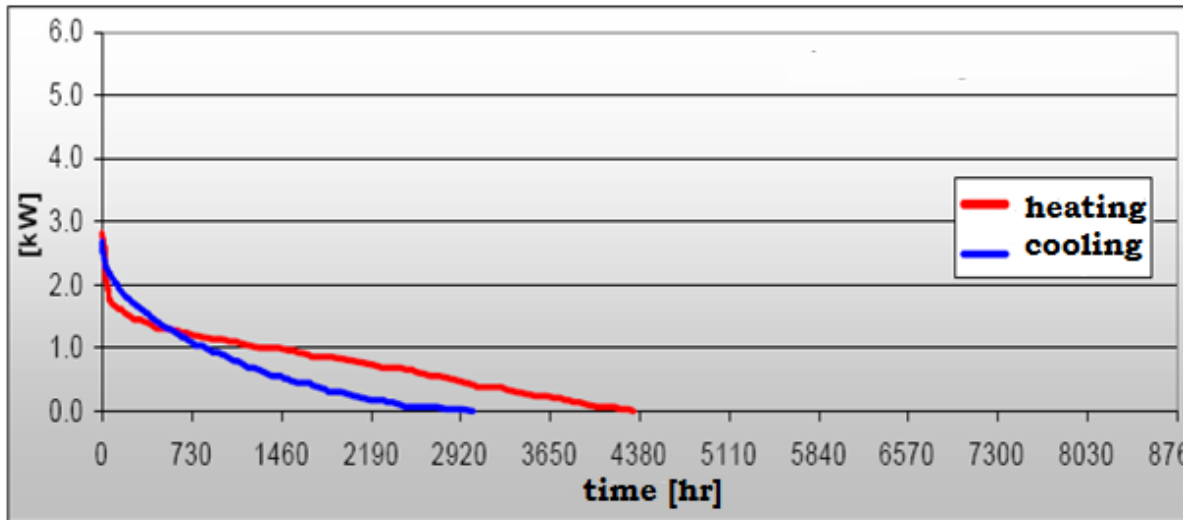


Figure 37. Example of an annual load duration curve constructed from Royal Haskoning data.

When the load duration curve for cooling is very flat (not a large peak on the left), the chances of imbalance when option 1 is chosen are smaller than when it is very steep.

The advantage of option 2 is that all cold stored in the ground by the heat pump system can be extracted in the cooling season, provided the rules outlined in the beginning of this section on the matching of the heat-to-cold ratio are taken into consideration when sizing the heat pump system. The disadvantage of this option is however that the variable costs of the reservoirs are at their maximum values.

When the control system is designed smart, and the cooling load duration curve permits it, there is a third option. This option involves sizing the reservoir such that they are as small as possible, without having the problems of imbalance. This can be done by providing a base load by the direct cooling from the underground reservoir, while the remainder is done by cooling machines.

Concluding remarks

All information given in this section provides some general picture of what is to be taken into consideration when a heating and cooling system coupled to geothermal energy storage is designed. It shows that there is a lot more to it than energy alone.

9 Discussion

In Chapters 4, 7 and 8 the results from the modelling work are given. In this chapter all results are summarized and discussed.

Detailed heat pump model

The results of the simulations of the Robur GAHP-AR reversible air source heat pump have shown that an up going trend is observed for the heating power when outdoor air temperatures rise, which corresponds to manufacturer data. There are some differences however when it comes to the exact trend. The manufacturer data show that from a certain outdoor air temperature on the heating power does not rise any longer, whereas the model predicts otherwise.

The fact that this sort of system behaviour is not captured in the model can have many causes. First of all there is the modelling of the internal heat exchange at pre-absorber, rectifier and internal heat exchanger CE, which is done using fixed temperature differences at the exchanger outlets and by solving the energy balances. It would have been more realistic to model this using the NTU method which is also used for the condenser and the evaporator. This has been tried but turned to lead to solver start-up and stability problems

Other important model simplifications included assuming a fixed ammonia concentration in the vapour at the generator outlet and a fixed temperature for the rectifier. The temperature (profile) inside the generator was unknown and had to be assumed. The same can be said about the saturation temperature of the strong solution. Also, the model assumes that there is no pressure drop inside any of the system components, which also deviates from the real situation. The model does show that the data provided by the manufacturer should be attainable physically.

Heating system model

The heating system model has been built to quantify the difference between different heating system concepts which serve to answer the question raised in Chapter 1: are gas absorption heat pumps a good means to lower the energy use for heating and cooling of buildings with a high ratio of heating demand to cooling demand?

The heating system model built consisted of several gas absorption heat pumps, combined with gas boilers and a buffer vessel. This system was set to meet a pre-determined heating profile of a nursing home. For the systems using geothermal storage, the amount of cold stored underground was also recorded. The model was set to simulate an entire year.

The results of this modelling exercise are given in Chapter 7. From these results it becomes apparent that all gas absorption heat pump heating systems have better seasonal system efficiency values than a heating system consisting of condensing gas-fired boilers. This efficiency advantage becomes less with rising building supply temperatures.



When ground source and water source systems are compared, the water source system shows a significantly better energetic heating performance than the ground source system. There are two main reasons that justify this difference. Firstly, the water source system has higher temperatures at its disposal at the heat pumps evaporator, which positively affects the efficiency of the heat pump cycle. Secondly, since the ground source system works with lower temperature differences, it requires much more pumping energy for a given amount of heat supplied to the evaporator than the water source system.

When looking at the heat-to-cold ratio of the systems using geothermal storage, the ground source system has a higher heat-to-cold ratio than the water source system. This is due to the fact that the temperature of the water supplied to the evaporator is lower than for the water source system, which means a lower evaporator duty and thereby a lower heat pump efficiency.

The heating system model does have its limitations. For example, the control system model imposes that as soon as a heat pump has shut down, no other heat pump is allowed to start up for the next two hours. This is sort of control systems are often employed in practice, but the chosen 'off-time' varies from situation to situation. The simulation time step also has quite a significant effect on the final modelling results. Choosing a smaller value will benefit the accuracy at the expense of the computational efficiency. Another factor that influences the accuracy of the results is the fact that the part-load behaviour of the gas-fired boilers and the ground pumps is not modelled in the present situation.

Match between heat pump and application

In Chapter 8 five different heating and cooling system concepts for meeting the demands of the reference nursing home are compared by looking at their annual expense of primary energy. The results show that using a heating and cooling system with gas absorption heat pumps coupled to heat and cold storage in aquifers in combination with gas-fired boilers and an air-cooled chiller results in the lowest primary energy use of all systems considered.

It is important to realize that although cooling is considered in Chapter 8, no cooling demand curve has been simulated to obtain results on the cooling energy use. The cooling energy use calculations are based on data from a manufacturer of cooling machines and on the energy requirements for the ground pumps calculated earlier in Appendix D. Then there is the COP of the vapour-compression heat pump. This COP is obtained from manufacturer data and is considered to be constant throughout the year. Also, no start-up losses are taken into account for the vapour compression heat pump, while these are taken into account for the gas absorption heat pump.

The most important message about Chapter 8 is that it is important to realize that there are differences between the modelling accuracy of the different system components and that these differences influence the final results.



10 Conclusion

The goal of the present project is to find out whether gas-fired heat pumps are a good means to lower the energy use for heating and cooling of buildings with a high ratio of heating demand to cooling demand. The project focussed on gas absorption heat pumps (GAHP's) and nursing homes.

Since little detailed information on these machines was available within Royal Haskoning, a thermodynamic model of a typical gas absorption heat pump for the built environment was developed in Matlab/Simulink. The results from this modelling have shown that the efficiency values given by its manufacturer are physically attainable.

In buildings, heating systems commonly consist of heat pumps, gas-fired boilers and a buffer vessel. A model of such a heating system using GAHP's was built. This heating system model was coupled to a derived heat demand profile of a typical 3000 m² nursing home. Three different heat sources were modelled for the heat pumps: outdoor air, geothermal heat from the ground (ground source) and geothermal heat from underground aquifers (water source).

The heating system model was run using these three different heat sources for a fixed building heating water supply temperature of 45 °C. Several heat pumps were put in parallel as to make sure they would be able to meet 50% of the peak heating demand. The remainder of the heating capacity was delivered by gas-fired boilers.

Simulations using the heating system model were performed for the NEN5060 climate year and the corresponding annual primary energy consumption of these systems was calculated. These simulations proved that the water source GAHP has the best energetic performance, followed by the ground source heat pump. The air source heat pump has the worst energetic performance.

To consider both heating and cooling, five different heating and cooling concepts were envisioned. All concepts make use of gas-fired boilers for backup heating. The first three concepts used air source, ground source and water source GAHP's, while the fourth used a water source vapour compression heat pump. Furthermore, the concepts using GAHP's also used an air-cooled chiller for supplementary cooling. The fifth system was a gas-fired boiler combined with an air-cooled chiller.

For all five different heating and cooling systems the annual primary energy expenses were calculated. The results showed that the heating and cooling concept using the gas absorption heat pump with the water source had the lowest primary energy consumption. The ground source gas absorption heat pump system also showed a favourable primary energy demand. The gas boiler only system performed the worst. It can thus be concluded that of the concepts considered for nursing homes a water source gas absorption heat pump, combined with gas-fired boilers and an air cooled chiller provides the best means to lower the buildings primary energy use for heating and cooling.





11 Recommendations

In Chapter 9 the results from the project were discussed. Also the limitations of the present models and the effects of these limitations on the final results were discussed. The present chapter will provide recommendations on how the models can be adapted to improve the accuracy of their results. First the detailed heat pump model will receive some attention, followed by the heating system model. The chapter ends with recommendations for improving the calculations relating to the match between the heat pump and the application.

Detailed heat pump model

The detailed heat pump model of the Robur GAHP-AR heat pump was based on limited information. As was shown in Chapter 4, the heating power predicted by the model does lie in the same range as the heating power given by the manufacturer, but the trend predicted by the model deviates from the trend as given by the manufacturer for varying outdoor temperatures. The model would really benefit from more information on the temperatures inside the system and from information on the control system employed.

Heating system model

There are also a number of notes to make on the heating system model. Firstly, the choice was made to model a flow rate controlled heating system. Although these systems are encountered in practice, they are relatively rare because of the higher installation costs. The model therefore gives good predictions only for a heating system employing a control system that is relatively rare. The model can of course be adapted to work with temperature control, but this did not have sufficient priority to carry it out within the framework of the present project.

Another important limitation of the heating system model is the coupling between supply and return temperatures. This arises because of the fact that the manufacturer data, on which the heat pump models are based, consist of fixed combinations of supply and return temperatures. Therefore the temperature decrease of the building heating water when it flows through the building is fixed.

The quality of the results from a model is always strongly dependent on the quality of its input. Because data on the heating demand profile of nursing homes was lacking, this profile was calculated using some rules of thumb obtained from the monitoring of office buildings. It is apparent that these rules of thumb may not apply to nursing homes, even if they are adapted to the situation using common sense. Therefore the modelling results would benefit from real-life data on the heating demand of a nursing home.

There are more components in the building heating system than just the heat pumps. Because this project focused on heat pumps and because the heat pumps are the most complicated components in the system, they received the most attention. The gas-fired boilers and ground



source pumps therefore were modelled in much less detail. Efforts were made to accurately estimate their efficiencies under normal load conditions. Part-load behaviour was not taken into account however. Incorporating part-load behaviour would benefit the accuracy of the heating system model predictions.

Another important recommendation given here is on the computational model efficiency of the heating system model. This computational efficiency is severely affected by the simulation step size. Since a smaller simulation step size means a slower calculation, but also a more accurate model, a trade-off is to be made here. It would be nice if ways were found to decrease computational time without affecting the accuracy of the calculations. Solutions should be looked for in the control system model, where the simulation time step is used to calculate how much heat pumps will be started up at a given moment.

Match between heat pump and application

When it comes to the heating system modelling results as discussed in Chapter 8, there are some limitations as well.

Firstly, not all interesting heating and cooling concepts were modelled and compared. The gas engine driven vapour-compression heat pump is the most important absentee. With its heat-to-cold ratio of about 1.9 and favourable energy efficiency values, it is expected that the gas engine driven heat pump will perform well in applications using much more heat than cold on a yearly basis. Furthermore it is able to decrease its heating power stepless to 25% of the maximum value. For this reason the contribution of a gas engine heat pump to the yearly heating energy supply is expected to be higher than for gas absorption heat pumps when the installed heating power is the same for both systems. This has a positive effect on the total SPF of the heating system, since the share of the gas-fired boilers decreases. Combined with the high efficiency values encountered in these gas engine heat pumps it is expected that they will be able to meet the requirements of nursing homes at low expenses of primary energy. It is therefore advised that the primary energy consumption of heating systems using gas engine heat pumps is predicted using the model described in Chapter 5, with some modifications. Only when the results from such simulations are known, it can be stated with reasonable certainty which heating and cooling system is best suited for meeting the demands of nursing homes.

A second limitation of the results of Chapter 8 lies in differences in the level of detail with which the various alternatives were modelled. While the modelling of heating systems using the gas absorption heat pumps was done quite thoroughly, the same cannot be said for the electrically driven vapour-compression heat pump and the air-cooled chiller. In order to make a better comparison possible, these systems should receive the same modelling attention as the gas absorption heat pumps.



Acknowledgements

For the past three months I have been given the chance to submerge myself into a topic which really interests me. I am grateful that Royal Haskoning has provided me with this opportunity. When I started I was familiar with heat pumps, since I have done two courses on this topic in the first year of my Master studies. I was not too familiar with the use of heat pumps in building systems, however. It has been a real learning experience for me and I am very content with my choice for this topic and this company.

I want to thank my supervisors for making this project possible.

My supervisor at the Delft University of Technology was Carlos Infante Ferreira. With his many years of experience in the refrigeration field he can with right be called an expert. This project would not have been what it is today without his support. I am glad he was able to make time for my project, while he was already very busy supervising other students. The appointments we had every two weeks have proven very valuable and I was always amazed by his preparation for these meetings. Even if I had only sent my report a few hours before, he had read the most important parts to make sure we had something to discuss. His critical look has done a lot to improve my project and my report. The desire to go to the heart of the topic and to understand everything about it is something we both have in common. This made our communication very easy.

My supervisor at Royal Haskoning was Maarten Rensen. A few minutes before my first day started we had our first meeting in the train to the office. By coincidence he recognized the internship assignment I was reading, after which he introduced himself to me. From the start our contact has been very pleasant. Once every week I went to Nijmegen to discuss the progress of my assignment with him. On the days I worked from the office in Rotterdam we had regular contact through chat, e-mail and telephone. Maarten has proven to be a very good supervisor: professional, knowledgeable, involved, friendly, supportive and demanding. He kept a close eye on my progress without giving me the feeling he was pushing me. He really wanted me to make the most out of this project and looking back I can say with certainty that he succeeded in doing so.

There are of course numerous people, inside and outside of the company that I should thank for their support. In fear of forgetting somebody important I will leave it to the names of my supervisors here. I do want to thank all people of the buildings services department in both Rotterdam and Nijmegen for their friendliness. From the first day I have felt at home within Royal Haskoning and I am sure I am going to miss some of my 'colleagues'.





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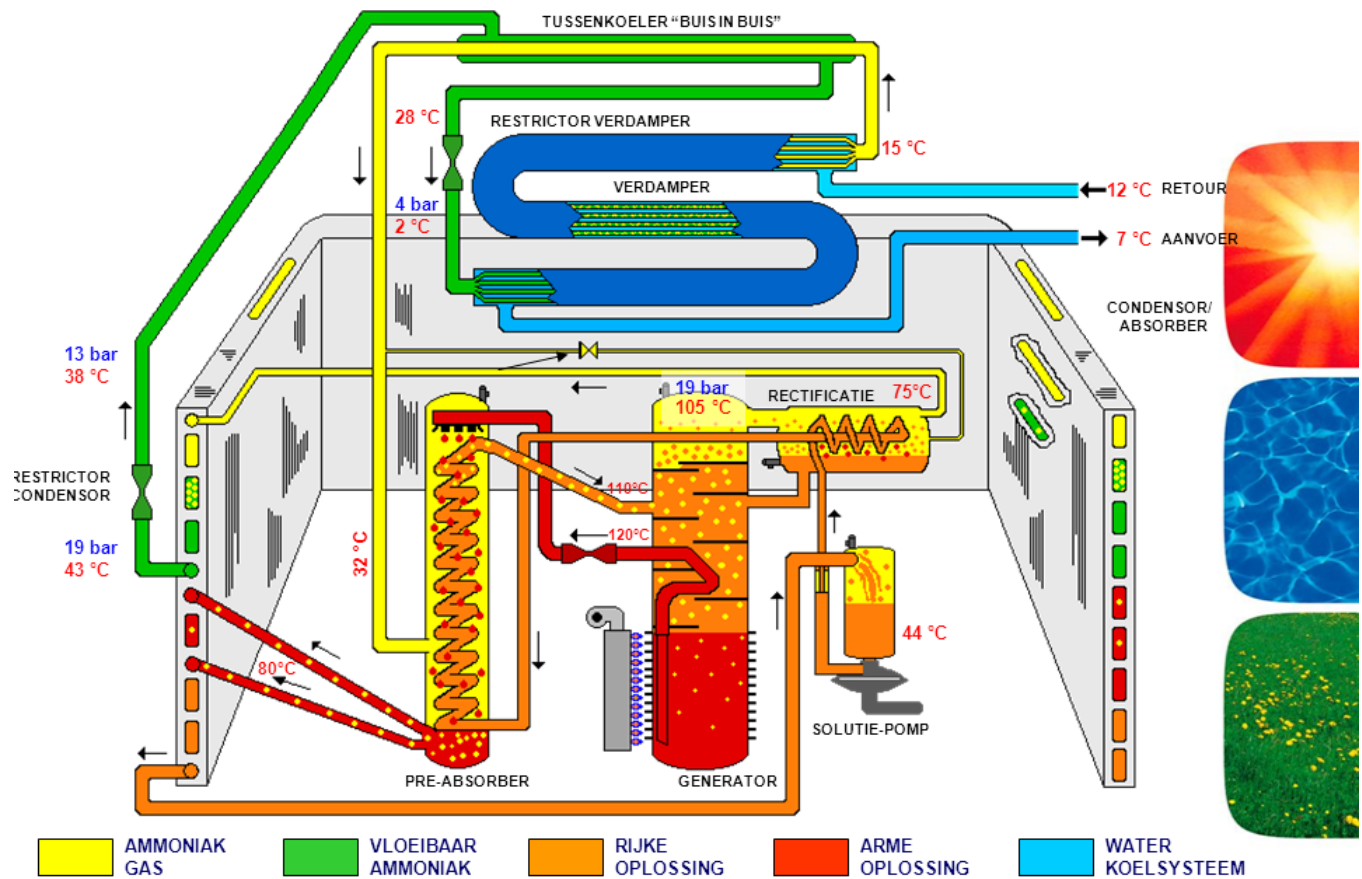


Appendices





Appendix A

Process flow diagram from Techneco



Appendix B

Adsorption

Like absorption heat pump systems, adsorption systems are also heat driven. Heat is used to force the refrigerant out of a solid adsorbent. The gaseous refrigerant condenses in a condenser, is expanded, evaporates in an evaporator at decreased pressure and is then adsorbed again, releasing more heat. The different phases encountered in an adsorption heat pump are shown in Figure 38.

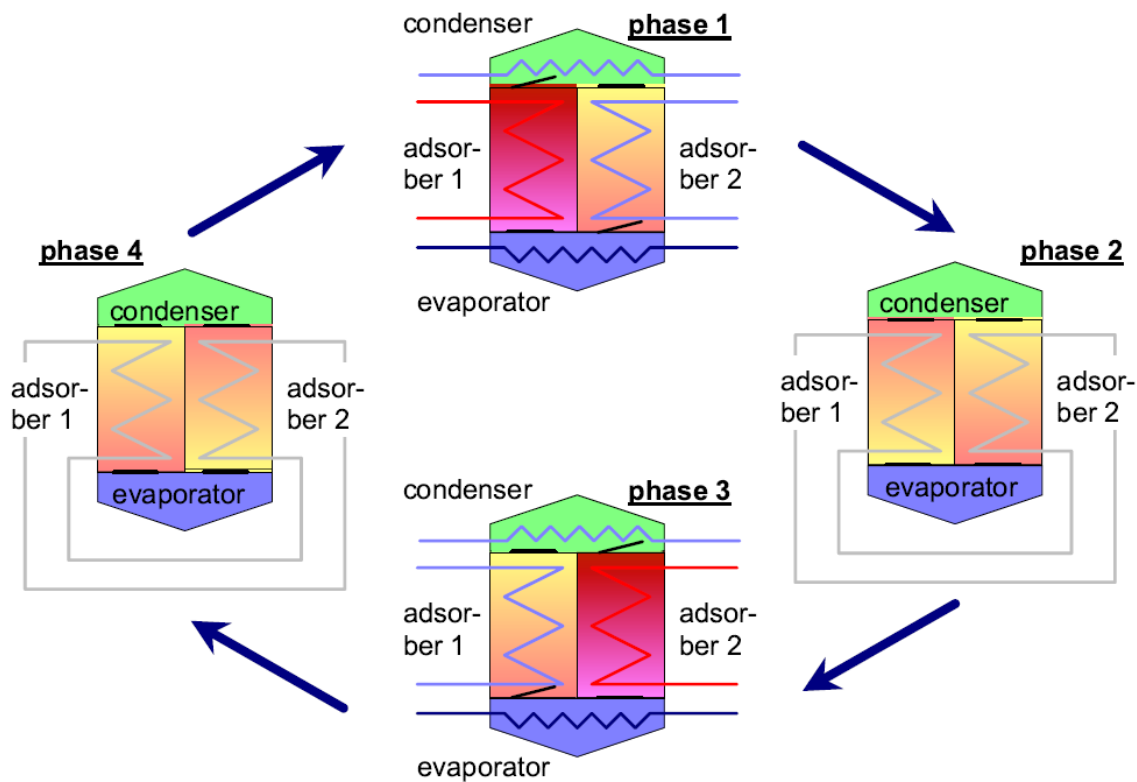


Figure 38. Graphic showing the different phases encountered in an adsorption heat pump cycle. (Podesser, 2003)

The phases shown in Figure 38 will be explained below.

Phase 1 – desorption

Refrigerant is driven off from the adsorbent through the use of hot water.

Phase 2 - condensation

The refrigerant vapour flows to the condenser, where it condenses, giving off heat which can be used in the building.

Phase 3 – evaporation

The liquid from the condenser passes through an expansion valve after which it enters the evaporator. There it evaporates at a lower temperature, taking heat from the heat source.

Phase 4 – adsorption

The vapour from the evaporator passes to the adsorber, where it is adsorbed on the adsorbent, releasing heat.

The main advantage of adsorption systems is that the heat used to drive the cycle can be of even lower temperature than the heat used to drive absorption cycles. Driving temperatures start at about 55 °C, compared to about 70 °C for absorption heat pumps.

The adsorption type heat pump has not yet seen much application in the built environment. The only manufacturer providing adsorption systems of reasonable sizes is the Japanese company Mayekawa (Mayekawa, 2011).



Appendix C

Heat pump efficiency definitions

Electricity (grid)-driven heat pump

For electricity driven heat pumps the coefficient of performance (*COP*) is used to measure the performance in heating mode. Different *COP* definitions exist. The simplest *COP* definition is the quotient of the heat delivered by the heat pump (*HP*) condenser to the electrical power required by the compressor:

$$COP_{HP} = \frac{\dot{Q}_{cond}}{\dot{E}_{HP,comp}^H} \quad (1.1.1)$$

There are also *COP* definitions which take all electrical power consumption by the heat pump system (*sys*) into account, except electrical resistance heating:

$$COP_{sys} = \frac{\dot{Q}_{cond}}{\dot{E}_{HP,comp}^H + \dot{E}_{aux}^H} \quad (1.1.2)$$

This *COP* includes the auxiliary power consumption by fans, control equipment and heat pump related pumps.

Another important efficiency ratio is the primary energy ratio (*PER*) of the heat pump in heating mode. It relates the heat delivered by the heat pump to the primary energy used to power the compressor:

$$PER_{HP} = \frac{\dot{Q}_{cond}}{\dot{E}_{HP,comp}^H / \eta_{el,grid}} \quad (1.1.3)$$

Where the electric efficiency for power generation to use in the heat pumps compressor is defined as:

$$\eta_{el,grid} = \frac{\dot{E}_{HP,comp}}{\dot{m}_{fuel,power_plant} LHV_{fuel}} \quad (1.1.4)$$



Because heat pumps often operate under part-load conditions instead of full power, a so-called seasonal performance ratio is defined. This ratio is very meaningful because it takes part-load, starts and stops, electrical resistance heating (*ERH*), defrosting and auxiliary power consumption into account during a certain period of time with predefined weather and heating demand. Its definition for the whole heat pump system during the heating season is:

$$SPF_{sys} = \frac{\int (\dot{Q}_{WH} + \dot{Q}_{SH}) \cdot dt}{\int (\dot{E}_{HP,comp}^H + \dot{E}_{ERH} + \dot{E}_{aux}^H) \cdot dt} \quad (1.1.5)$$

In the numerator the amount of heat used for water heating (*WH*) and space heating (*SH*) are represented. The seasonal performance factor for the heating season is sometimes called the heating season performance factor (*HSPF*). Care should be taken however, because there exists a definition of the heating season performance factor which mixes British and SI units:

$$HSPF_{sys} = \frac{\int (\dot{Q}_{WH} + \dot{Q}_{SH}) \cdot dt \text{ [Btu]}}{\int (\dot{E}_{HP,comp}^H + \dot{E}_{ERH} + \dot{E}_{aux}^H) \cdot dt \text{ [Wh]}} \quad (1.1.6)$$

The HSPF definition of equation (1.1.6) gives much higher efficiency values for the heating season than the definition of equation (1.1.5) because 1 Wh = 3.412 Btu.

For the heating season also the primary seasonal performance factor for the whole system can be calculated. It equals the primary energy ratio for the whole system and its definition is given in equation(1.1.7):

$$SPF_{prim} = \frac{\int (\dot{Q}_{WH} + \dot{Q}_{SH}) \cdot dt}{\int \left(\left[\dot{E}_{HP,comp}^H + \dot{E}_{ERH} + \dot{E}_{aux}^H \right] / \eta_{el,grid} \right) \cdot dt} \quad (1.1.7)$$

Most heat pumps can also function as cooling machines. The condenser of the heat pump then becomes the evaporator of the cooling machine and the evaporator of the heat pump becomes the condenser. In order to evaluate its performance in the cooling cycle often the energy efficiency ratio (*EER*) is used. This energy efficiency ratio is nothing more than the COP in cooling mode. It is defined as:



$$EER_{HP} = \frac{\dot{Q}_{evap}}{\dot{E}_{HP,comp}^C} \quad (1.1.8)$$

Again there is an efficiency ratio going by the same name but mixing up British units and SI units. It gives much higher energy efficiency ratio values:

$$EER_{HP} = \frac{\dot{Q}_{evap} [\text{Btu/hr}]}{\dot{E}_{HP,comp}^C [\text{W}]} \quad (1.1.9)$$

For the cooling cycle a seasonal performance ratio is defined as well. Its definition is very comparable to the definition of the heating season performance factor. Mixing British units and SI units we get:

$$SEER_{sys} = \frac{\int \dot{Q}_{evap} \cdot dt [\text{Btu}]}{\int (\dot{E}_{HP,comp}^C + \dot{E}_{aux}^C) \cdot dt [\text{Wh}]} \quad (1.1.10)$$

Eurovent efficiency certification

In order to introduce a common definition for the efficiency of heat pumps the Eurovent Certification Committee has introduced the European Seasonal Energy Efficiency Ratio (*ESEER*) for the cooling mode of a reversible heat pump. This efficiency definition uses the energy efficiency ratio (in SI units) for the entire heat pump system at different load conditions.

$$EER_{sys} (\% \text{ of total load}) = \frac{\dot{Q}_{evap}}{\dot{E}_{HP,comp}^C + \dot{E}_{aux}} \quad (1.1.11)$$

Using weighting factors (*A, B, C* and *D*) these values are used to calculate the *ESEER*:

$$\begin{aligned} ESEER_{sys} = & A \cdot EER(100\% \text{ load}) + B \cdot EER(75\% \text{ load}) \\ & + C \cdot EER(50\% \text{ load}) + D \cdot EER(25\% \text{ load}) \end{aligned} \quad (1.1.12)$$

In equation (1.1.12) the sum of the weighting factors *A, B, C* and *D* is unity. Their values are chosen by Eurovent.



Gas engine heat pump

For a heat pump employing a gas engine to drive a vapour-compression cycle the situation is a bit different. In this case the heat entering the building is obtained from the condenser as well as from waste heat of the gas engine itself. The gas utilization efficiency of the combination of heat pump and gas engine and the PER are again equal:

$$GUE_{HP\&GE} = PER_{HP\&GE} = \frac{\dot{Q}_{cond} + \dot{Q}_{GE_used}}{\dot{m}_{gas,GE} LHV_{gas} - \dot{E}_{aux} / \eta_{el,GE}} \quad (1.1.13)$$

In this equation $\dot{Q}_{cond}$ is the amount of heat delivered by the heat pump condenser, while $\dot{Q}_{GE_used}$ is the amount of waste heat from the gas engine used for heating the building. Note that the energy use for auxiliaries is not taken into account in this GUE definition. In a gas engine heat pump system the electricity used by the system is produced by the gas engine itself. This means converting this electricity use to gas consumption by the gas engine and subtracting it from the total amount of gas used by the gas engine. The electric efficiency of the gas engine is defined as:

$$\eta_{el,GE} = \frac{\dot{E}_{GE}}{\dot{m}_{gas} LHV_{gas}} \quad (1.1.14)$$

The seasonal performance factor for the system differs from the primary energy ratio because it takes the energy use of auxiliaries into account. Furthermore it does not use the heat from the condenser and the useful heat from the gas engine in the numerator but the amount of heat used in the *building* for hot water and space heating. This is, of course, the same if heat losses to the environment in the absorber and condenser are neglected:

$$SPF_{sys} = SPF_{prim} = \frac{\int (\dot{Q}_{WH} + \dot{Q}_{SH}) \cdot dt}{\int (\dot{m}_{gas,GE} + \dot{m}_{gas,boiler}) LHV_{gas} \cdot dt} \quad (1.1.15)$$

Because gas engine still produces waste heat when the heat pump is operating in cooling mode, gas engine heat pump systems are capable of providing heating and cooling at the same time.



Appendix D

Estimation of required pump power for ground source and water source heat pumps

In order to estimate the power required for the pumps which pump the heat from the aquifer source to the sink or through the vertical heat exchanger, estimations for pressure losses inside the system have to be made. These estimations were performed in consultation with Royal Haskoning's heat and cold storage expert Joep Woldring. (Woldring, 2012)

Water source

The pressure losses inside a real system depend on the specific conditions and equipment used. General guidelines for the pressure losses inside water source systems using heat and cold storage in underground aquifers are given in Table 14.

Table 14. Pressure losses inside a heating system using heat and cold storage inside aquifers.

Pressure loss [in m water]	
Friction losses	
Building side HEX	about 8
Pipes	about 10
Injection and extraction losses	
Extraction well	5 to 10
Injection well	10 to 15

Adding up the pressure losses in Table 14 we get a pressure loss of about 40 m of water, equal to about 4 bar. The real pressure loss inside the system also depends on flow velocity, but this is neglected in this simplified approach.

Assuming the hot source is at 15 °C all year long and the temperature of the water leaving the heat pump is 45 °C, the evaporator duty can be calculated as explained in section 5.2.1. This gives:

$$\dot{Q}_{\text{evaporator,WS}} = 16.87 \text{ kW}$$

Neglecting heat losses, this evaporator duty is equal to the duty of the building side heat exchanger (often abbreviated TSA).

Assuming all four heat pumps are on, the total duty becomes 67.5 kW. Using the specific heat capacity and density of water from Table 7 and a temperature difference of 8 K for the water passing the TSA we get for the volumetric flow rate:



$$\dot{V}_{\text{water,WS}} = \frac{\dot{Q}_{\text{TSA,WS}}}{c_{p,w} \rho_w \Delta T_{\text{TSA,WS}}} = 7.33 \text{ m}^3 / \text{hr}$$

Using Figure 39 from pump manufacturer Grundfoss, the SP 17-4 pump is chosen for this application.

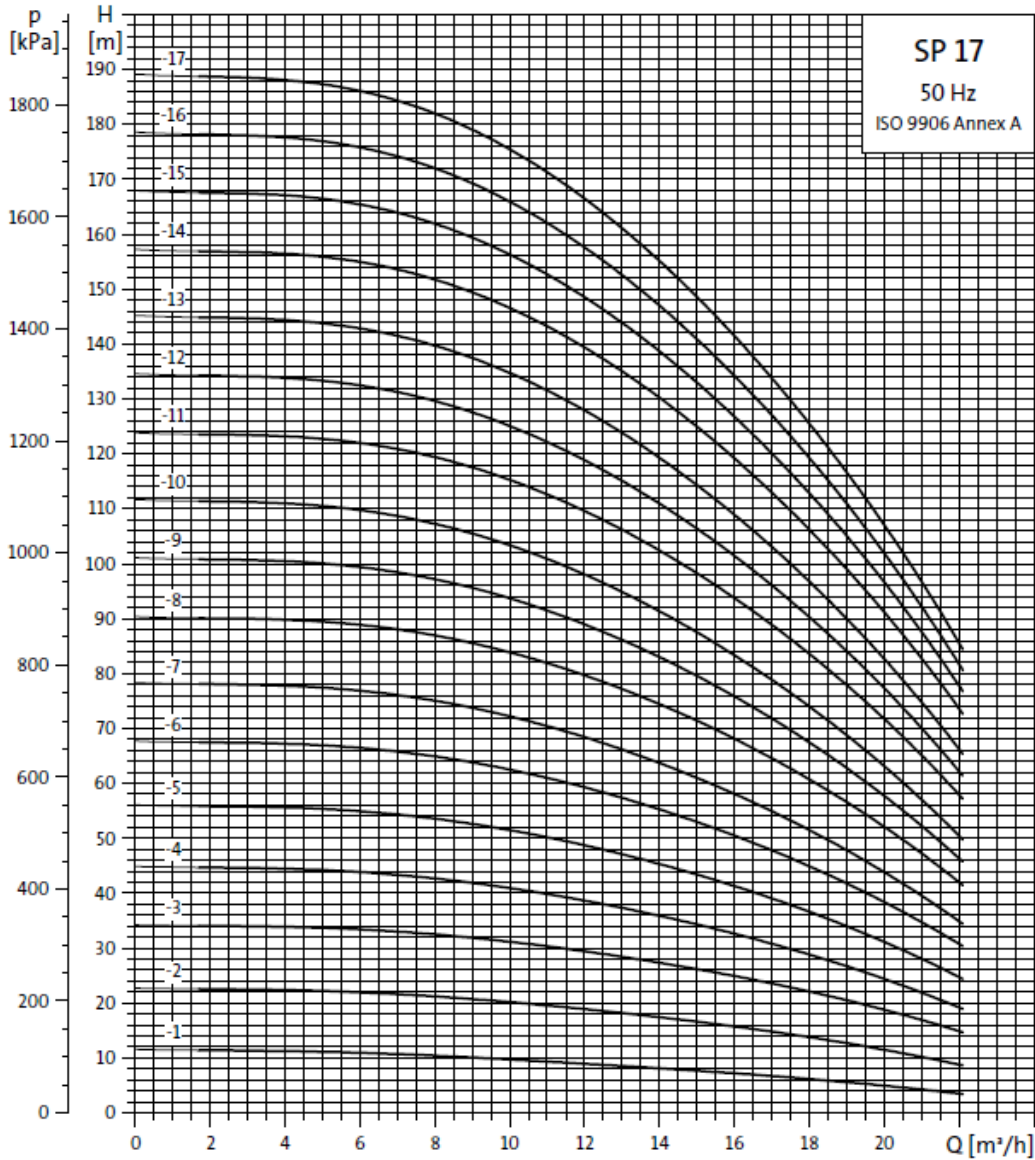


Figure 39. Pump characteristics for several submerged Grundfoss pumps.

The required pump work is read from Figure 40 to be 1.5 kW. From this number the ratio of pump work to TSA duty is calculated:

$$E_{\text{ratio_pumps,WS}} = \frac{1.5}{67.5} = 0.022 \text{ kW}_e / \text{kW}_{th}$$

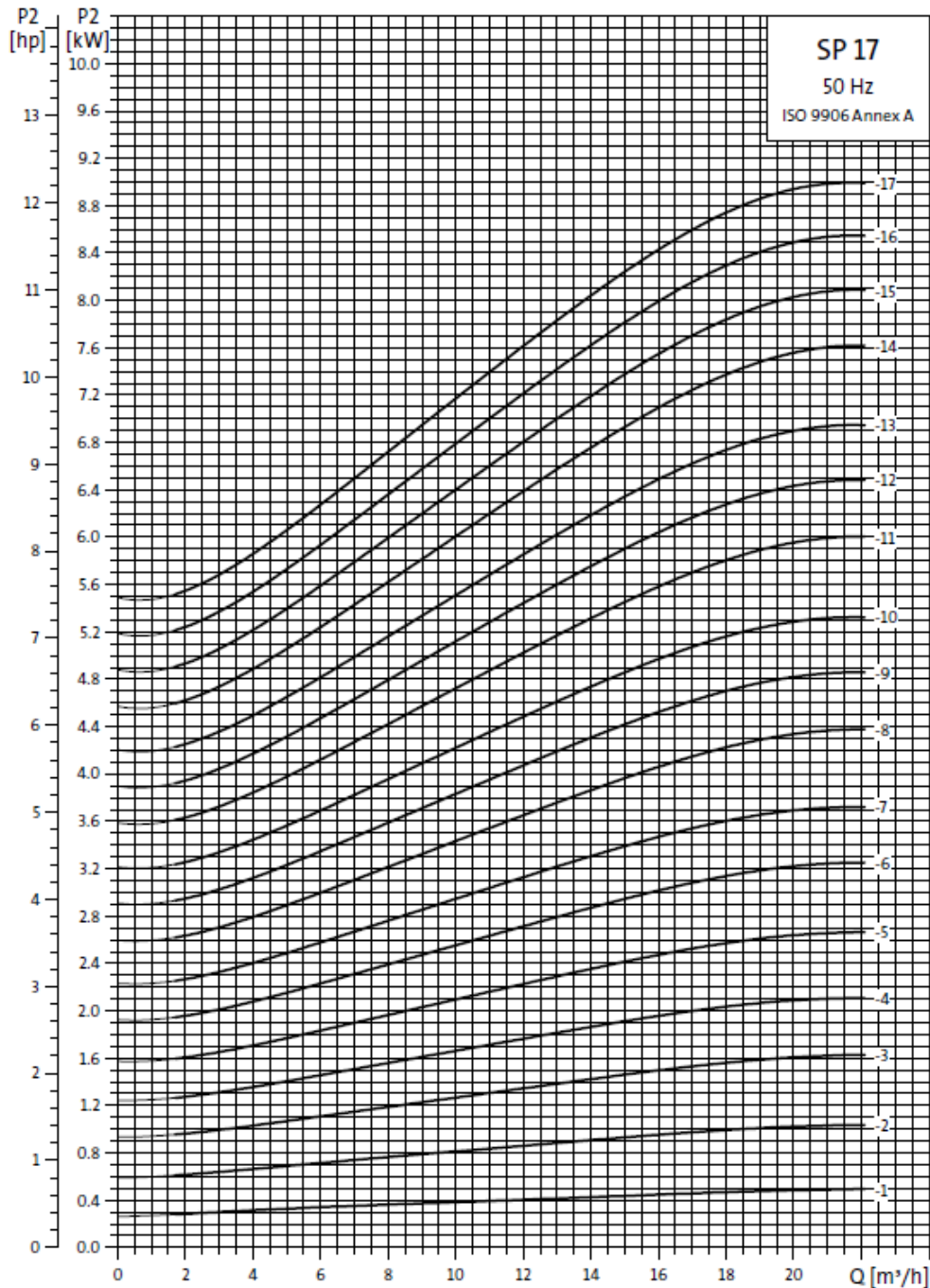


Figure 40. Power curve for several submerged pumps of manufacturer Grundfoss.

Ground source

The approach for the ground source heat pump is very similar. It is known that the pressure loss of these systems is generally quite somewhat higher than the pressure loss of aquifer systems. Taking an average source temperature of 7 °C and 45 °C as the temperature of the water leaving the heat pump, we get for the evaporator duty:

$$\dot{Q}_{\text{evaporator},GS} = 15.01 \text{ kW}$$

Since there are again four heat pumps employed, the duty of the TSA becomes: 60.04 kW. The temperature decrease of the water across the TSA is about 4 K for ground-source systems. The water flow rate now becomes:

$$\dot{V}_{\text{water},GS} = \frac{\dot{Q}_{TSA,GS}}{c_{p,w} \rho_w \Delta T_{TSA,GS}} = 13.03 \text{ m}^3 / \text{hr}$$

Figure 39 shows that the SP 17-6 pump fits this application. Using Figure 40 once again the power requirement is obtained to be 2.8 kW. The ratio of pump work to TSA duty for the ground source heat pump thus becomes:

$$E_{\text{ratio_pumps},GS} = \frac{2.8}{60.04} = 0.047 \text{ kW}_e / \text{kW}_{th}$$



Appendix E

Calculation of the start-up losses

The gas absorption heat pump is notorious for its long start-up times. Information on the typical start-up behaviour of a Robur gas absorption heat pump was obtained from the manufacturer. The simplified picture for the GAHP-AR model looks as follows (Figure 41):

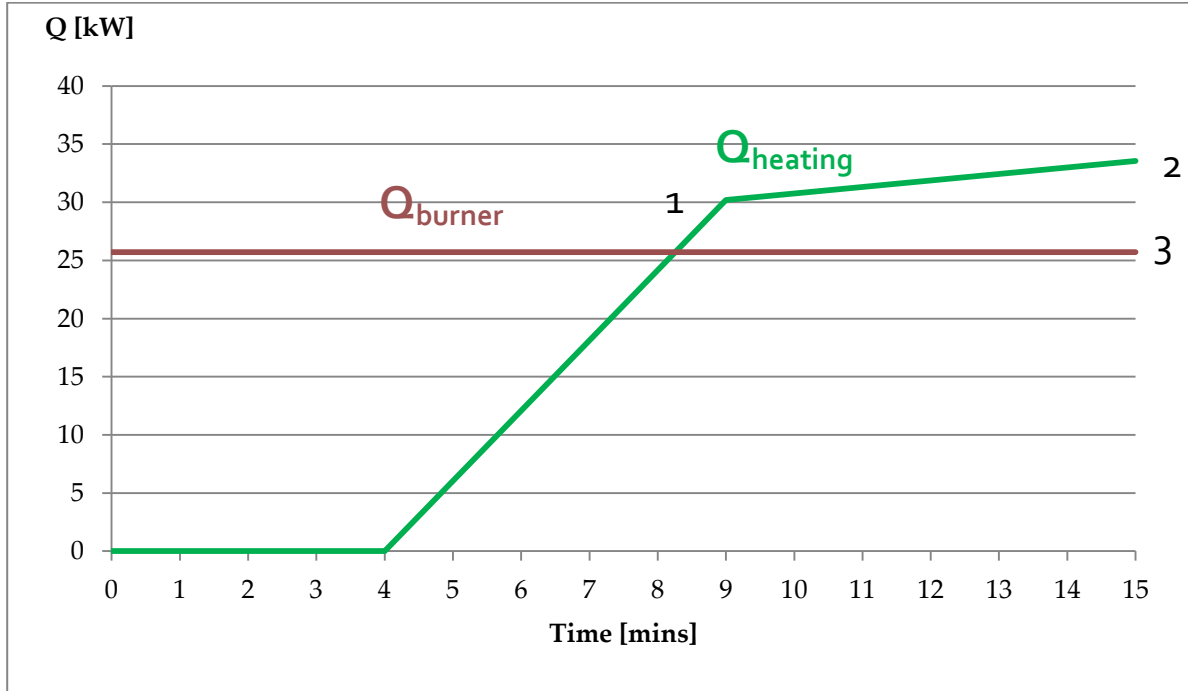


Figure 41. Typical start-up behaviour for the Robur GAHP-AR reversible air source heat pump. The actual start-up time may be as long as 20 minutes. In this figure the more common value of 15 minutes is chosen. The value of the heating power has been corrected with the correction factor 0.95 which is used throughout this report. See section 5.2.1 for more information on this correction factor.

The start-up losses are now calculated as the heat input minus the heat output:

$$Q_{gas,startup} [kWh] = \int_{t=0}^{15 \text{ mins}} \dot{Q}_{burner} dt - \int_{t=0}^{15 \text{ mins}} \dot{Q}_{heating} dt = 6.425 - 4.4435 = 1.9815 \text{ kWh}$$

For the interested reader, the values at points 1, 2 and 3 are respectively: 30.18, 33.54 and 25.7 kW.

Although the start-up behaviour and losses may be slightly different for the ground source and water source heat pumps, the same start-up loss is used for these systems.

Differences arise when the electrical power consumption during start-up is taken into account. For air source heat pumps this is merely the auxiliary power requirement, integrated over the time span during which the heat pump is starting up:

$$E_{el,startup,AR}[kWh] = \int_{t=0}^{15 \text{ mins}} \dot{W}_{el,aux,HP} dt = (0.9 \text{ kW}) \cdot (0.25 \text{ hr}) = 0.225 \text{ kWh}_e$$

For the ground-source heat pump we have:

$$\begin{aligned} E_{el,startup,GS}[kWh] &= \int_{t=0}^{15 \text{ mins}} \left(\dot{W}_{el,aux,HP} + E_{ratio_pumps,GS} \cdot \dot{Q}_{evaporator,GS} \right) dt \\ &= (0.47 \text{ kW} + 0.047 \cdot 15.0095 \text{ kW}) \cdot (0.25 \text{ hr}) = 0.294 \text{ kWh}_e \end{aligned}$$

And for the water source heat pump this becomes:

$$\begin{aligned} E_{el,startup,WS}[kWh] &= \int_{t=0}^{15 \text{ mins}} \left(\dot{W}_{el,aux,HP,WS} + E_{ratio_pumps,WS} \cdot \dot{Q}_{evaporator,WS} \right) dt \\ &= (0.47 \text{ kW} + 0.022 \cdot 16.8748 \text{ kW}) \cdot (0.25 \text{ hr}) = 0.210 \text{ kWh}_e \end{aligned}$$

All these energy amounts can be added to yield the primary energy used during start-up:

$$Q_{startup} = Q_{gas,startup} + E_{el,startup} / \eta_{el,grid}$$

Appendix F

Eurovent conditions for Carrier 30RB 162 kW air-cooled chiller

Deellast prestaties

Met de snelle stijging van de energiekosten en de zorg om het effect van elektriciteitsproductie op het milieu, is het energieverbruik van airconditioning apparatuur een belangrijk punt van discussie geworden. De energie efficiency van een vloeistofkoelmachine bij vollast is zelden representatief voor de werkelijke prestaties van de units, omdat een koelmachine gemiddeld minder dan 5% van de bedrijfstijd in vollast werkt.

De warmtelast van een gebouw is afhankelijk van meerdere factoren, zoals de buitenluchttemperatuur, de oriëntatie (zon/schaduw) en de bezettingsgraad.

Het verdient daarom de voorkeur om de seizoens energie efficiency te hanteren, berekend bij verschillende bedrijfscondities die representatief zijn voor het belastingspatroon.

ESEER (EUROVENT)

Met de ESEER (European seasonal energy efficiency ratio) kan de gemiddelde energie efficiency bij deellast worden uitgedrukt. Dit op basis van vier, door Eurovent gedefinieerde bedrijfscondities. De ESEER is de gemiddelde waarde van energie efficiency verhoudingen (EER) bij verschillende bedrijfscondities, gewogen door de bedrijfstijd.

ESEER (European seasonal energy efficiency ratio)

Belasting, %	Buitenluchttemperatuur, °C	Energie efficiency	Bedrijfstijd, %
100	35	EER ₁	3
75	30	EER ₂	33
50	25	EER ₃	41
25	20	EER ₄	23
ESEER = EER ₁ x 3% + EER ₂ x 33% + EER ₃ x 41% + EER ₄ x 23%			
Opmerking: constante wateruitredetemperatuur = 7°C			

Deellastprestaties conform Eurovent

30RB 162-262 "B" units met optie 280 (shell-and-tube warmtewisselaar) en 30RB 302-802 units					
30RB	Load %	CAP kW	Unit kW	EER kW/kW	ESEER kW/kW
162	100	162,7	54,1	3,01	
	75	122,0	33,9	3,60	
	50	81,3	19,7	4,14	
	25	40,7	9,3	4,39	3,99
182	100	181,3	60,2	3,01	
	75	136,0	38,6	3,53	
	50	90,7	22,5	4,02	
	25	45,3	10,3	4,38	3,91
202	100	197,2	71,3	2,77	
	75	147,9	43,2	3,42	
	50	98,6	25,5	3,86	
	25	49,3	12,1	4,08	3,74
232	100	227,3	73,9	3,07	
	75	170,4	46,1	3,70	
	50	113,6	25,9	4,39	
	25	56,8	11,8	4,80	4,22
262	100	270,9	99,8	2,72	
	75	203,2	61,0	3,33	
	50	135,5	33,2	4,08	
	25	67,7	15,3	4,44	3,87
302	100	297,8	106,1	2,81	
	75	223,4	63,2	3,54	
	50	148,9	36,6	4,07	
	25	74,5	16,4	4,54	3,96
342	100	331,8	122,1	2,72	
	75	248,9	72,0	3,45	
	50	165,9	40,4	4,11	
	25	83,0	18,2	4,55	3,95
372	100	367,0	129,5	2,83	
	75	275,2	74,7	3,69	



Appendix G

Details on the detailed gas absorption heat pump model of Chapter 3

1 Pre-absorber (GAX-absorber)

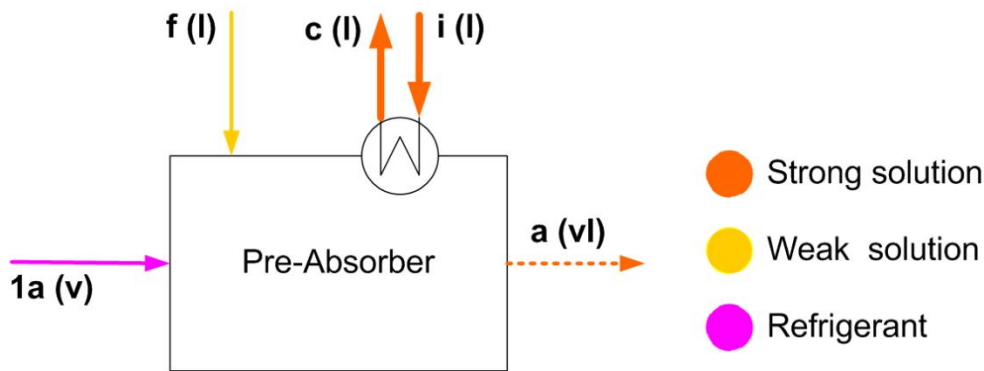


Figure 42. Modelling layout: Pre-absorber.

Inputs

$$x_{weak}, \dot{m}_R, P_{evap}, h_{1a}, h_f, h_i, P_{cond}$$

Parameters

$$T_{sat} = 50 \text{ } ^\circ\text{C}^{(1)}, \Delta T_{c-f} = 10 \text{ K}^{(2)}$$

Outputs (7)

$$x_{strong}, \dot{m}_{strong}, \dot{m}_{weak}, T_f, T_c, h_c, h_a, T_{sat}$$

NOTES ON PARAMETER VALUES:

- 1) The saturation temperature was chosen such that it is 5 K above the temperature of the building heating water leaving the condenser ($T_{w,building,in}$). This is done to make sure that there is a sufficiently large driving force in the absorber, where the water is heated from $T_{w,building,out}$ to $T_{w,abs,out}$.

The temperature difference between streams $\underline{c}$ and $\underline{f}$ is taken from a flow chart provided by Techneco (see Appendix A).

In the pre-absorber the weak solution from the generator (stream **f**) is brought into contact with the ammonia vapour coming from the internal heat exchanger (stream **1a**). A part of it is absorbed into the liquid phase, while the remaining ammonia vapour leaves with the solution as dispersed bubbles (inside stream **a**). The enthalpy of the solution with its dispersed ammonia bubbles leaving the pre-absorber is calculated with an energy balance.

Assumptions

- Fixed saturation temperature (T_{sat}) for the strong solution determines the overall concentration of stream **a**
- Fixed temperature difference (ΔT_{c-f}) between streams **c** and **f**
- No heat loss (adiabatic pre-absorber)

Equations

In the pre-absorber module first the saturation concentration of the strong solution leaving the absorber (stream **g** in Figure 9) is calculated:

$$x_{strong} = x(P_{evap}, T_{sat}) \quad (1.1.1)$$

In equation (1.1.1) the saturation temperature T_{sat} is a parameter which is set in the model parameter window. It is chosen such that there exists a temperature difference of at least 5 K between the strong solution leaving the absorber and the water entering the absorber. With the concentration of the strong solution known, the overall mass balance and ammonia mass balance can be set up. These yield respectively:

$$\begin{aligned} \dot{m}_{strong} &= \dot{m}_R + \dot{m}_{weak} \\ \dot{m}_{strong} x_{strong} &= \dot{m}_R + \dot{m}_{weak} x_{weak} \end{aligned}$$

Combining both equations and rewriting in terms of the unknowns gives:

$$\dot{m}_{strong} = \frac{\dot{m}_R (1 - x_{weak})}{x_{strong} - x_{weak}} \quad (1.1.2)$$

$$\dot{m}_{weak} = \dot{m}_{strong} - \dot{m}_R \quad (1.1.3)$$



Next the temperatures and enthalpies of some entering and leaving streams are calculated:

$$T_f = T(x_{weak}, P_{evap}, h_f) \quad (1.1.4)$$

Assuming a fixed temperature difference ΔT_{c-f} between streams c and f, which exchange heat, we get:

$$T_c = T_f - \Delta T_{c-f} \quad (1.1.5)$$

And thus the enthalpy of stream c becomes:

$$h_c = h(x_{strong}, P_{cond}, T_c) \quad (1.1.6)$$

With all these enthalpies known, the enthalpy of stream a can be calculated from the pre-absorber energy balance:

$$h_a = \frac{\dot{m}_R h_{1a} + \dot{m}_{weak} h_f + \dot{m}_{strong} h_i - \dot{m}_{strong} h_c}{\dot{m}_{strong}} \quad (1.1.7)$$

2 Absorber

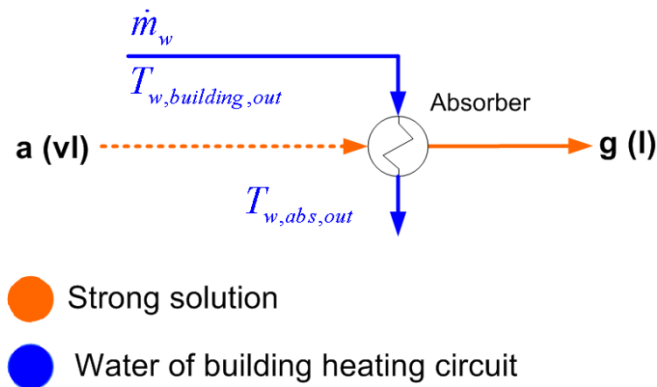


Figure 43. Modelling layout: Absorber.

Inputs

$$x_{strong}, T_{w,building,out} = 35^\circ\text{C}, \dot{m}_w, \dot{m}_{strong}, h_a, P_{evap}$$

Parameters

$$T_{sat} = 50^\circ\text{C}^{(1)}, c_{p,w} = 4.1797 \text{ kJ/kg} \cdot \text{K}^{(2)}$$

$$\Delta T_{sub} = 5 \text{ K}^{(3)}$$

Outputs (4)

$$T_g, h_g, \dot{Q}_{solution}, T_{w,abs,out}$$

NOTES ON PARAMETER VALUES

- 1) The saturation temperature was chosen such that it is 5 K above the temperature of the building heating water leaving the condenser ($T_{w,building,in}$). This is done to make sure that there is a sufficiently large driving force in the absorber, where the water is heated from $T_{w,building,out}$ to $T_{w,abs,out}$. The saturation temperature, together with the evaporating pressure, determines the concentration of the strong solution which is calculated in the pre-absorber module (see equation 1.1.1).
- 2) The constant specific heat capacity of water was obtained from the RefProp thermodynamic libraries. It is the average between the specific heat capacity at 35 °C and at 45 °C.
- 3) A sub cooling of 5 K was assumed for the solution leaving the absorber to make sure that all ammonia is absorbed into the liquid phase.

Assumptions

- Fixed sub-cooling of the strong solution at the absorber outlet (ΔT_{sub})
- No heat loss (adiabatic absorber)
- Constant specific heat capacity for the building heating water ($c_{p,w}$) at the given temperature range

Equations

In the absorber the building heating water is heated. The ammonia-water solution from the pre-absorber with ammonia bubbles inside is cooled down by the thermal contact with the building heating water. It is assumed that the ammonia-water solution leaves the absorber sub-cooled:

$$T_g = T_{sat} - \Delta T_{sub} \quad (1.2.1)$$



With the temperature known, the enthalpy of the strong solution leaving the absorber can be calculated:

$$h_g = h(x_{strong}, P_{evap}, T_g) \quad (1.2.2)$$

Since the enthalpy and mass flow rate of the incoming strong solution are known and mass conservation is valid for the absorber, the amount of heat released can be calculated:

$$\dot{Q}_{solution} = \dot{m}_{strong} (h_a - h_g) \quad (1.2.3)$$

The temperature of the building heating water leaving the absorber can now be calculated from the energy balance on the absorber:

$$T_{w,abs,out} = T_{w,building,out} + \frac{\dot{Q}_{solution}}{\dot{m}_w c_{p,w}} \quad (1.2.4)$$

3 Solution pump

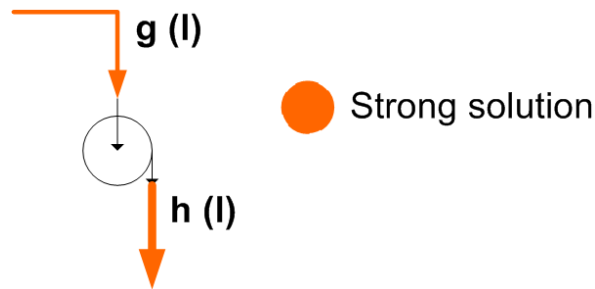


Figure 44. Modelling layout: Solution pump.

Inputs

$$h_g, \dot{m}_{strong}, x_{strong}, P_{evap}, P_{cond}$$

Parameters

$$\eta_{is,sol_pump} = 0.50^{(1)}, \eta_{mech} = 0.98^{(2)}$$

Outputs (4)

$$\dot{W}_{sol_pump_ideal}, v_1, \dot{W}_{sol_pump}, h_h$$

NOTES ON PARAMETER VALUES

- 1) The isentropic pump efficiency was taken from literature (Velázquez and Best, 2002).
- 2) The mechanical efficiency (shaft to pump) is generally high, therefore a value of 0.98 was assumed.

Assumptions

- The strong solution entering the solution pump is incompressible
- Isentropic pump efficiency does not vary with pressure ratio
- No heat losses (adiabatic pump)

Equations

The solution pump takes the strong solution from the absorber from evaporator pressure to condenser pressure. For an ideal pump the power required is given by:

$$\dot{W}_{sol_pump_ideal} = \dot{m}_{strong} v_g (P_{cond} - P_{evap}) \quad (1.3.1)$$

In the previous equation the specific volume of the strong solution is calculated using the conditions at the pump inlet:

$$v_g = v(x_{strong}, P_{evap}, h_g) \quad (1.3.2)$$



Using the pump efficiency parameters, the real pump power can be computed from the ideal pump power:

$$\dot{W}_{sol_pump} = \frac{\dot{W}_{sol_pump_ideal}}{\eta_{is,sol_pump} \eta_{mech}} \quad (1.3.3)$$

The enthalpy of the strong solution after the pump is obtained from the energy balance around the pump:

$$h_h = h_g + \frac{(\dot{W}_{sol_pump} \cdot \eta_{mech})}{\dot{m}_{strong}} \quad (1.3.4)$$

Equation (1.3.4) assumes that all heat generated by the fact that the pumping occurs non-isentropic is taken up by the strong solution. This originates from the assumption that the pumping occurs adiabatic.

4 Generator

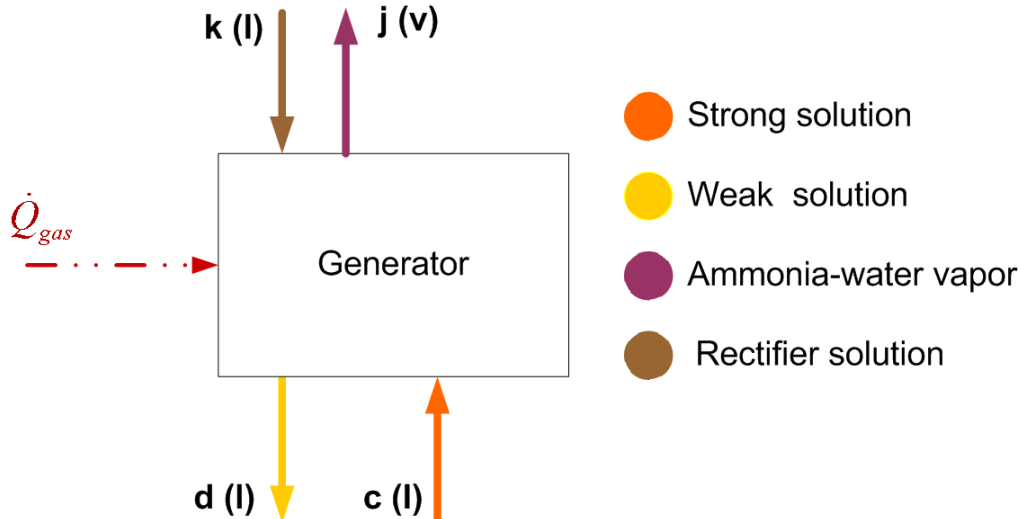


Figure 45. Modelling layout: Generator.

Inputs

$$\dot{m}_k, \dot{m}_{weak}, \dot{m}_{strong}, h_k, h_c, P_{cond}$$

Parameters

$$T_{gen} = 150 \text{ }^{\circ}\text{C}^{(1)}, y_j = 0.96^{(2)}, \dot{Q}_{gas, fed} = 25.7 \text{ kW}^{(3)}, \\ \eta_{burner} = 0.94^{(4)}, \Delta T_{gen, weak} = 15 \text{ K}^{(5)}, \Delta T_{gen, vap} = 15 \text{ K}^{(6)}$$

Outputs (6)

$$x_{weak}, h_j, h_d, \dot{m}_j, T_{gen, weak, out}, T_{gen, vap, out}$$

NOTES ON PARAMETER VALUES:

- 1) For choosing a suitable generator temperature, a literature source was consulted. The article under consideration is on the thermodynamic modelling of a gas-fired ammonia-water GAX absorption **chiller** (Chua et al., 2002). The chiller modelled in this article is the Arkla ACC 3600. This chiller is from the Arkla-Servel company, of which the gas conditioning division was acquired by Robur in 1991 (Robur, 2012). In the article a generator temperature of 145 °C was used. However, in order for the heat pump to run at temperatures of -20°C a higher generator temperature is required to avoid problems with the concentration of the weak solution. These problems arise since the concentration of the strong solution decreases at decreasing outdoor temperatures (= decreasing evaporating pressure). Therefore the weak solution concentration has to be low enough to avoid a strong solution concentration lower than the weak solution concentration. A means to achieve this is increasing the generator temperature. Therefore a generator temperature of 150°C was taken.
- 2) The ammonia mass fraction in the gaseous phase leaving the generator is based on the same article. In the article 0.92 is used, in the model 0.96 is used to avoid heat transfer problems in the rectifier at high outdoor temperatures. This problem can arise due to the fact that the refrigerant flow rate increases with increasing outdoor temperatures, while the flow rate of the strong solution decreases. This can mean that the amount of heat that has to be taken away in the rectifier is so large that the temperature of the leaving strong solution $\dot{m}_j$ will have to be larger than the rectifier temperature. Since this is non-physical, the higher ammonia mass fraction was used.
- 3) For the thermal input to the heat pump the manufacturer data from the Robur GAHP-AR model was taken from the Techneco website (Techneco, 2011a).



- 4) For the burner efficiency a value of 94% was assumed. This value is reasonable for a non-condensing burner system as employed in the GAHP-AR.
- 5) Generators are designed to have a temperature gradient in order to enhance the separation efficiency. From Figure 2 in the modelling article (Chua et al., 2002) it can be seen that the generator has a higher temperature at the bottom, which is where the burner is located. Since the weak solution leaves the generator at about half-height (see Appendix A) its temperature is read from Figure 46 to be 15 K lower than the temperature at the bottom of the generator.
- 6) The vapour phase leaves the generator at a temperature also 15K colder relative to the bottom of the generator. The gas has a temperature higher than the sub-cooled solution it emerges from due to bad heat transfer between the gas bubbles and the solution.

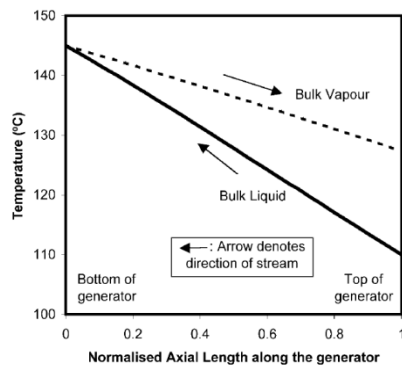


Fig. 2. Local temperature profiles of the streams within the generator.

Figure 46. From the article on thermodynamic modelling of a GAX absorption chiller (Chua et al., 2002).

Assumptions

- The generator consists of three zones each with their own uniform temperature
- The weak solution leaving the generator (stream d) is saturated at the generator pressure and generator temperature
- Constant heat input rate with natural gas ($\dot{Q}_{gas}$)
- Constant ammonia vapour fraction in vapour stream leaving the generator (stream j)

Equations

In the generator the strong solution is heated, releasing ammonia and water vapour. The resulting weak solution is sent back to the pre-absorber (stream d), the vapour is carried up to the rectifier where it is purified further (stream j). In the calculation process first the concentration of the solution leaving the generator is calculated. It is given by:

$$x_{weak} = x(P_{cond}, T_{gen}) \quad (1.4.1)$$

The generator does not have a uniform temperature throughout. In order to account for this phenomenon, three temperature zones are distinguished: bottom, weak solution outlet and vapour (top). These zones are labelled T_{gen} , $T_{gen,weak,out}$ and $T_{gen,vap,out}$ respectively. With T_{gen} given, the temperature of the leaving weak solution and the leaving vapour can be calculated:

$$T_{gen,weak,out} = T_{gen} - \Delta T_{gen,weak} \quad (1.4.2)$$

$$T_{gen,vap,out} = T_{gen} - \Delta T_{gen,vap} \quad (1.4.3)$$

With all temperatures known and the generator pressure equal to the condenser pressure the enthalpies of the leaving streams can be calculated:

$$h_j = h(y_j, P_{cond}, T_{gen,vap,out}) \quad (1.4.4)$$

$$h_d = h(x_{weak}, P_{cond}, T_{gen,weak,out}) \quad (1.4.5)$$

The energy balance for the generator can now be used to calculate the vapour mass flow flowing to the rectifier:

$$\dot{m}_j = \frac{\dot{m}_k h_k + \dot{m}_{strong} h_c - \dot{m}_{weak} h_d + \dot{Q}_{gas, fed} \eta_{burner}}{h_j} \quad (1.4.6)$$



5 Rectifier

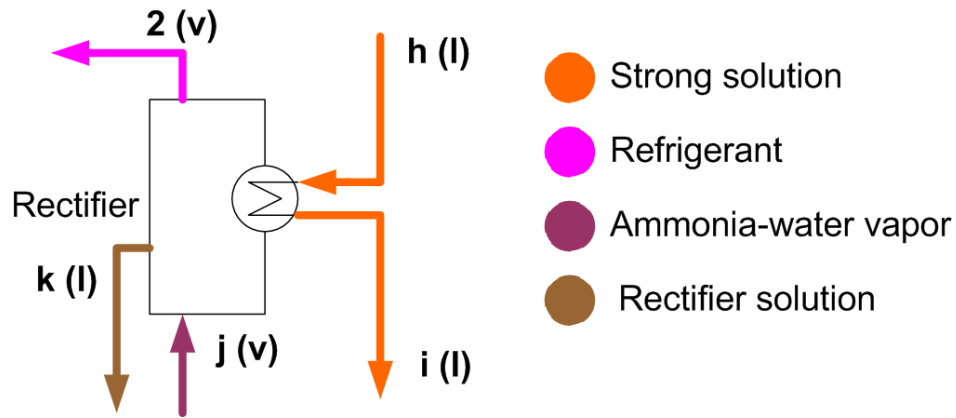


Figure 47. Modelling layout: Rectifier.

Inputs

$$\dot{m}_{strong}, \dot{m}_j, h_j, h_h, P_{cond}$$

Parameters

$$T_{rect} = 76^\circ\text{C}^{(1)}, y_j = 0.96, \Delta T_{rect,vap} = 20\text{K}^{(2)}$$

Outputs (6)

$$x_k, \dot{m}_k, \dot{m}_R, h_2, h_k, h_i$$

Notes on parameter values

- 1) The rectifier temperature of 76 °C is chosen with reference to the research article on the thermodynamic modelling of an Arkla absorption chiller (Chua et al., 2002).
- 2) It turns out that the vapour leaving the rectifier and flowing to the condenser is not at rectifier temperature but somewhat higher. The temperature difference is taken from (Chua et al., 2002) where the ammonia vapour leaving the rectifier is 20 K above rectifier temperature.

Assumptions

- The overall composition of the rectifier's contents is represented by the composition of its inlet stream j
- The composition of the reflux stream k depends on the vapour-liquid equilibrium at given temperature and pressure

- Pure ammonia vapour (stream 2) at rectifier pressure and a temperature above the rectifier temperature leaves the rectifier (perfect separation)
- Fixed rectifier temperature (T_{rect})

Equations

In the rectifier the vapour stream from the generator (stream j) is further purified at the generator pressure but at a lower temperature. To this end the vapour is partially condensed and the resulting liquid stream is carried back to the generator (reflux stream k). The partial condensation is performed by releasing heat to the strong solution stream coming from the solution pump (stream h).

The equilibrium composition of the reflux stream can be calculated from the entering composition and the conditions in the rectifier:

$$x_k = x(y_j, P_{cond}, T_{rect}) \quad (1.5.1)$$

For the rectifier also overall and ammonia mass balances can be set up. This gives:

$$\begin{aligned} \dot{m}_j &= \dot{m}_k + \dot{m}_R \\ \dot{m}_j y_j &= \dot{m}_k x_k + \dot{m}_R \end{aligned}$$

Rewriting yields:

$$\dot{m}_k = \frac{\dot{m}_j (1 - y_j)}{1 - x_k} \quad (1.5.2)$$

$$\dot{m}_R = \dot{m}_j - \dot{m}_k \quad (1.5.3)$$

The enthalpies of the streams leaving the rectifier are known from the conditions inside the rectifier. For the leaving ammonia vapour:

$$h_2 = h(P_{cond}, T_{rect} + \Delta T_{rect, vap}) \quad (1.5.4)$$

And for the reflux stream:

$$h_k = (x_k, P_{cond}, T_{rect}) \quad (1.5.5)$$

The leaving enthalpy of the strong solution flowing through the rectifier's heat exchanger is obtained from the energy balance:

$$h_i = \frac{\dot{m}_j h_j + \dot{m}_{strong} h_h - \dot{m}_R h_2 - \dot{m}_k h_k}{\dot{m}_{strong}} \quad (1.5.6)$$



6 Condenser

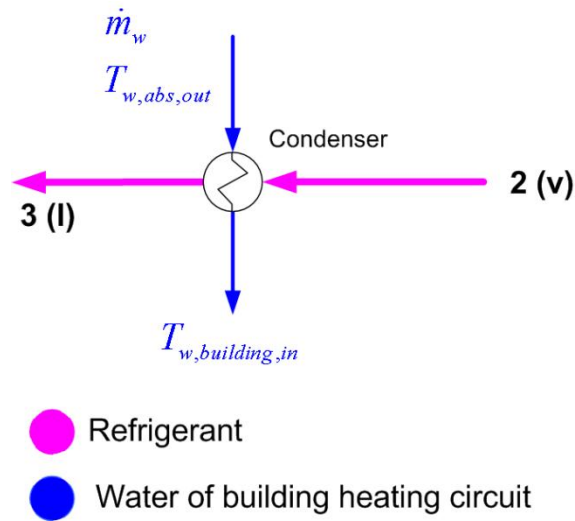


Figure 48. Modelling layout: Condenser.

Inputs

$$T_{w,abs,out}, T_{w,building,in} = 45^\circ\text{C}, h_2, \dot{m}_R$$

Parameters

$$c_{p,w} = 4.1797 \text{ kJ/kg} \cdot \text{K}, UA_c = 2.73 \text{ kW/K}^{(1)}$$

Outputs (6)

$$T_3, h_3, \dot{Q}_{cond}, P_{cond}, NTU_c, \dot{m}_w$$

Notes on parameter values

- 1) The UA_c value was estimated using the NTU-method (see section 3.2.2). Fixing the temperature of the water leaving the absorber to 40°C , the temperature of the water leaving the condenser to 45°C , the and the condensation temperature to 50°C , equations (1.6.1) and (1.6.2) can be solved for UA_c . The heating power is assumed to be equal to the heating power at 25°C outdoor temperature and the water flow rate used in the calculations is calculated accordingly.

Assumptions

- Constant condenser temperature

- No sub-cooling at condenser outlet
- No heat losses (adiabatic condenser)
- Constant specific heat capacity of the building heating water at the temperature range considered
- Constant UA-value (NTU-method can be used)

Equations

The building heating water pre-heated in the absorber is led to the condenser where it is heated up towards the desired temperature. The condensation temperature depends on the required temperature levels for the building heating water and the water mass flow rate. It is calculated using the NTU method:

$$T_3 = \frac{T_{w,building,in} - T_{w,abs,out} \cdot e^{-NTU_c}}{1 - e^{-NTU_c}} \quad (1.6.1)$$

where:

$$NTU_c = \frac{UA_c}{\dot{m}_w c_{p,w}} \quad (1.6.2)$$

With the condensation temperature known, the condensation pressure can be calculated:

$$P_{cond} = P(T_3, q = 0) \quad (1.6.3)$$

It is furthermore assumed that there is no sub-cooling of ammonia at the condenser outlet:

$$h_3 = h(T_3, q = 0) \quad (1.6.4)$$

As the refrigerant flow rate is known, the condenser duty can be calculated:

$$\dot{Q}_{cond} = \dot{m}_R (h_2 - h_3) \quad (1.6.5)$$

The water flow rate needed for the energy balance to close now follows:

$$\dot{m}_w = \frac{\dot{Q}_{cond}}{c_{p,w} (T_{w,building,in} - T_{w,abs,out})} \quad (1.6.6)$$



7 Internal heat exchanger

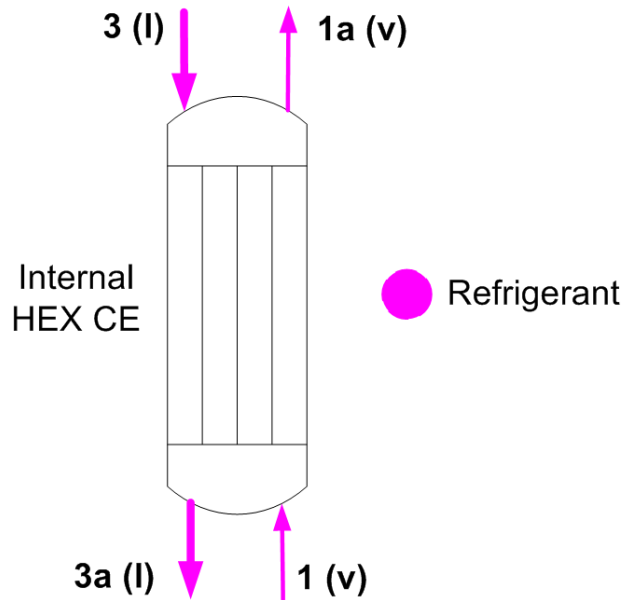


Figure 49. Modelling layout: Internal heat exchanger.

Inputs

$$P_{\text{evap}}, T_3, h_3, h_1$$

Parameters

$$\Delta T_{1a-3} = 20 \text{ K}^{(1)}$$

Outputs (3)

$$T_{1a}, h_{1a}, h_{3a}$$

Notes on parameter values

- 1) The temperature difference at the hot gas outlet side of the heat exchanger was taken to be the same as used in the research article (Chua et al., 2002)

Assumptions

- No heat losses (adiabatic heat exchanger)
- Fixed temperature difference between vapour outlet (stream 1a) and condensate inlet (stream 3) for the countercurrent heat exchanger

Equations

In the internal heat exchanger between condenser and evaporator (CE) the liquid ammonia from the condenser (stream **3**) is pre-cooled before it is expanded across the expansion valve. For computational efficiency reasons a constant temperature difference between the incoming liquid (stream **3**) and outgoing vapour (stream **1a**) is assumed instead of using the (iterative) NTU method:

$$T_{1a} = T_3 - \Delta T_{1a-3} \quad (1.7.1)$$

Because the temperature and pressure of stream **1a** are now known, its enthalpy can be calculated:

$$h_{1a} = h(P_{\text{evap}}, T_{1a}) \quad (1.7.2)$$

Noting that the refrigerant flow rate drops out, the energy balance on the adiabatic heat exchanger now yields the enthalpy of the leaving condensate stream:

$$h_{3a} = h_3 + h_1 - h_{1a} \quad (1.7.3)$$

8 Expansion valves

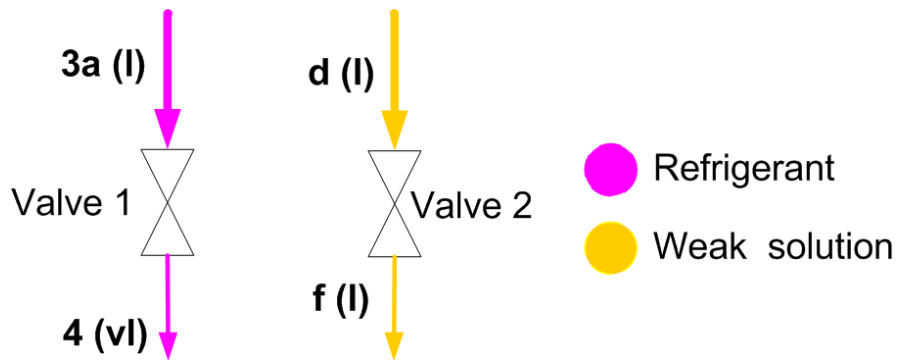
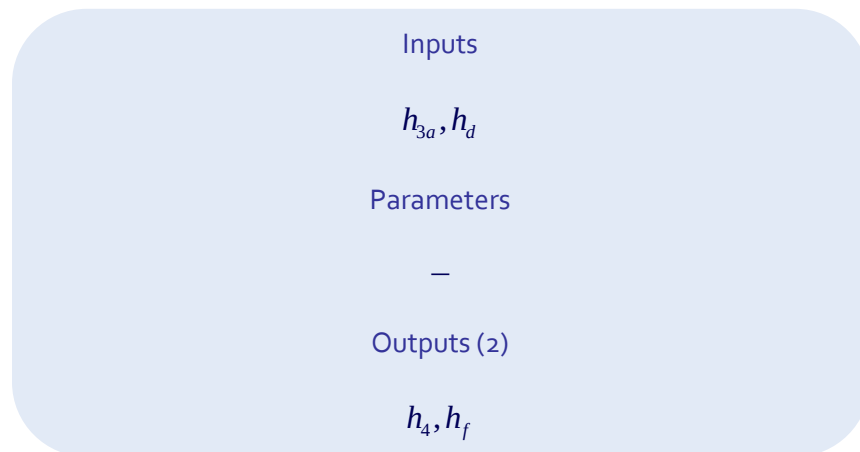


Figure 50. Modelling layout: Expansion valves.



Assumptions

- Adiabatic expansion valves (isenthalpic expansion)

Equations

Isenthalpic pressure reduction for expansion valve 1 gives:

$$h_4 = h_{3a} \quad (1.8.1)$$

And for valve 2:

$$h_f = h_d \quad (1.8.2)$$

9 Evaporator

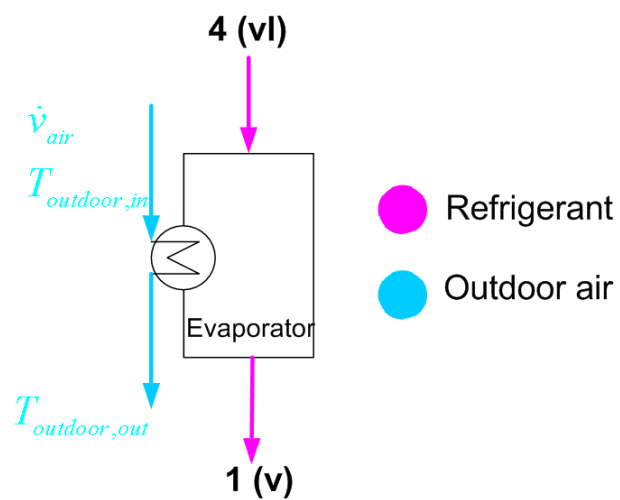


Figure 51. Modelling layout: Evaporator.

Inputs

$$h_4, \dot{m}_R, T_{outdoor,in}$$

Parameters

$$UA_e = 2.76 \text{ kW/K}^{(1)}, P_{air} = 1.01325 \text{ bar}^{(2)}, \dot{v}_{air} = 3.33 \text{ m}^3 / \text{s}^{(3)}$$

Outputs (9)

$$c_{p,air}, \rho_{air}, \dot{m}_{air}, NTU_e, T_1, h_1, P_{evap}, \dot{Q}_{evap}, T_{outdoor,out}$$

Notes on the parameter values:

- 1) The UA_e value was estimated using the NTU-method (see section 3.2.2). Fixing the temperature of the outdoor air entering the heat exchanger to 25°C, the temperature of the leaving air to 20°C and the evaporating temperature to 15°C, equations (1.9.4) and (1.9.5) can be solved for UA_e .
- 2) For the pressure of the outdoor air the standard value of 1 atm was chosen.
- 3) The volumetric flow rate of air passing through the evaporator coils was obtained from Robur heat pump supplier Techneco from Delft (Beekman, 2012).

The evaporator is the component where heat enters the heat pump system. Common heat sources include outdoor air, water from underground aquifers and the ground itself.

Assumptions:

- Constant UA-value (NTU-method can be used)
- Constant evaporating temperature (pure ammonia, no pressure drop inside evaporator coils)
- No superheating at the evaporator outlet
- Constant specific heat capacity for the air
- Constant air flow rate over the evaporator coils



Equations

First the properties of the air are calculated with respect to inlet conditions. The specific heat capacity:

$$c_{p,air} = c_p(P_{air}, T_{outdoor,in}) \quad (1.9.1)$$

And the density:

$$\rho_{air} = \rho(P_{air}, T_{outdoor,in}) \quad (1.9.2)$$

Using the volumetric volume flow rate of air the mass flow rate is calculated:

$$\dot{m}_{air} = \rho_{air} \dot{V}_{air} \quad (1.9.3)$$

The number of transfer units (NTU) is calculated next, the subscript 'e' refers to 'evaporator':

$$NTU_e = \frac{(UA)_e}{\dot{m}_{air} c_{p,air}} \quad (1.9.4)$$

From the number of transfer units the evaporator temperature can be obtained:

$$T_1 = \frac{T_{outdoor,out} - T_{outdoor,in} e^{-NTU_e}}{1 - e^{-NTU_e}} \quad (1.9.5)$$

This T_1 is equal to T_4 , since stream 4 is a two-phase vapour-liquid mixture of ammonia at evaporator pressure. It should be noted that $T_{outdoor,out}$ is unknown initially and therefore it is found through iteration. Furthermore it is important that the temperatures used in equation (1.9.5) are expressed in Kelvin, whereas degrees Centigrade are used as a unit in the rest of the equations. With the evaporation temperature known, the evaporator pressure and the enthalpy of the saturated ammonia vapour at the evaporator outlet can be calculated:

$$P_{evap} = P_{sat}(T_1, q=1) \quad (1.9.6)$$

$$h_1 = h(T_1, q=1) \quad (1.9.7)$$



Because the enthalpy at the evaporator inlet is an input to the calculation, as is the refrigerant mass flow rate, the evaporator duty can be calculated next:

$$\dot{Q}_{\text{evap}} = \dot{m}_R (h_1 - h_4) \quad (1.9.8)$$

The evaporator energy balance yields an expression for $T_{\text{outdoor,out}}$ which can be used in the iteration process described earlier:

$$T_{\text{outdoor,out}} = T_{\text{outdoor,in}} - \frac{\dot{Q}_{\text{evap}}}{\dot{m}_{\text{air}} c_{p,\text{air}}} \quad (1.9.9)$$



Appendix H

Refrigerant properties – technical aspects

The working fluid in a heat pump is often called 'refrigerant'. Not every substance is suitable to be a refrigerant. In this section the technical aspects of refrigerants are discussed. At the end of this section it should be clear why one substance is better suited to be a refrigerant in a certain application than the other.

Evaporating pressure

In a heat pump system the working fluid is circulated in order to transport heat from a cold place to a warmer place. In order to make this possible it evaporates at low temperature, taking heat from the surroundings and it condenses at a higher temperature. At the evaporator the system pressure is at its lowest value, while it reaches its maximum value in the condenser. In order to make sure that there is no leakage of air into the system, the working fluid is chosen such that it has a saturation pressure above atmospheric pressure in the evaporator. If the fluid meets this requirement without needing excessive pressures in the condenser to meet the desired temperatures there, it is suitable to be a refrigerant. The evaporating temperature at atmospheric pressure for four common refrigerants is displayed in Table 15. In this table R410A and R407C are zeotropic mixtures, while R134a and R717 (ammonia) are pure fluids (Dincer, 2010).

It should be noted that the principles outlined below do not apply to supercritical refrigeration cycles, like cycles employing CO₂ as the working fluid.

Table 15. Normal boiling points for four common refrigerants.

Refrigerant	Normal boiling point [°C]
R410A	-51.4
R407C	-36.6
R134a	-26.1
R717	-33.3

In this section various refrigerant properties are compared using an ideal heat pump cycle in which the evaporator operates at -3 °C and the condenser at 45 °C. Compression is isentropic and expansion isenthalpic. See Table 16.



Table 16. Ideal heat pump system used for calculation of various refrigerant properties.

T_{evap} [°C]	-3
T_{cond} [°C]	45
Superheating [K]	0
Subcooling [K]	0
Isentropic compression	
Isenthalpic expansion	

System capital costs

The choice of a refrigerant is always a trade-off between capital costs and efficiency. Two important variables determining the capital costs are the volumetric heating capacity and the condenser pressure. These are explained below.

Volumetric heating capacity

An important refrigerant property that determines the capital costs of the heat pump system is its volumetric heating capacity ($\phi_{v,heating}$). This is defined as the amount of heat given off in the condenser per unit refrigerant volume:

$$\phi_{v,heating} = \rho_{cond,in} \Delta h_{vl} \quad (1.1.1)$$

where

$\rho_{cond,in}$ is the density of the refrigerant as it enters the condenser [kg/m³]

Δh_{vl} is the condensation enthalpy of the refrigerant [kJ/kg]

A large volumetric heating capacity means that a smaller compressor can be chosen. The size of the piping can be smaller as well. All this results in a smaller, lighter and more compact system with lower capital costs. The volumetric cooling capacity can be calculated as well, using the evaporation enthalpy instead of the condensation enthalpy. (Podesser, 2003)

Condenser pressure

In a heat pump system the condensation pressure is determined by the desired temperature of the hot water it should deliver to the building. Each refrigerant has its own relation between condensation temperature and pressure and therefore the pressures required to reach a certain condensation temperature vary from one substance to another. The higher the condensation pressure, the higher the capital costs of the system and the higher the system weight becomes. (Dincer, 2010)



System compactness and costs - Four refrigerants compared

As an example the properties of four different refrigerants are compared below in Figure 52 for the heat pump system of Table 16.

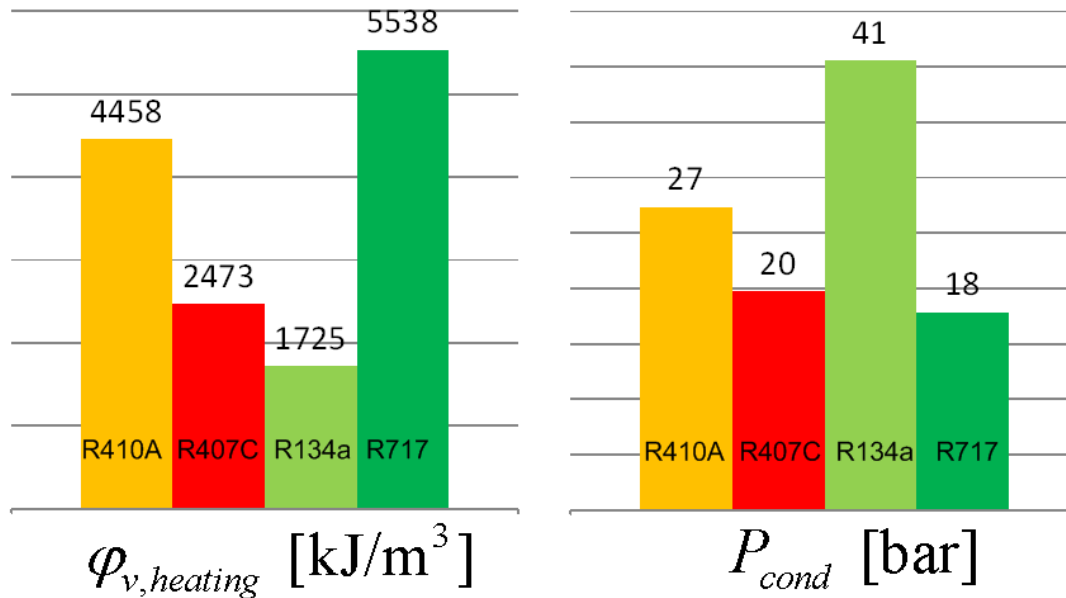


Figure 52. Refrigerant properties that determine system weight and capital costs for four different refrigerants using the heat pump system of Table 16.

It becomes clear from Figure 52 that ammonia (R717) is a very good refrigerant when it is important that the system is compact and low cost. It should be noted however that ammonia is toxic at low concentrations, contrary to the other three refrigerants which have a low toxicity. Therefore, additional demands on system tightness might partially mitigate its costs advantages. Furthermore, due to its chemical properties ammonia is not compatible with copper. Therefore, steel is often used in ammonia systems. Ammonia systems thus require some additional attention compared systems using one of the three HFC (hydrofluorocarbon) refrigerants mentioned in Figure 52 (Dincer, 2010).

Energy costs – system energetic efficiency

The running costs of a heat pump system are not only determined by its capital costs but in a large part also by its running costs. These running costs are determined by maintenance costs and energy costs. This sub section deals exclusively with energy costs. To this end a few refrigerant properties are discussed that have an influence on the system COP. At the end again four refrigerants are compared to quantify the effects of these properties.

Ratio of specific heat capacity to latent heat of vaporization

In the expansion device the saturated or sub-cooled refrigerant coming from the condenser is expanded into the two-phase region. The refrigerant thus partially vaporizes upon expansion. The

energy losses in this expansion device are governed by the ratio of the latent heat of vaporization to the specific heat capacity. The higher this ratio, the lower the losses are in the expansion device and thus the higher the energetic efficiency. (Podesser, 2003)

Transport properties

To keep the required surface area and thus the capital costs for the heat exchangers low, the heat transfer coefficient has to be high. Choosing an appropriate refrigerant can help in achieving this. A high thermal conductivity for example means a higher heat transfer coefficient. The effect of viscosity and density are more subtle. This is because while a low viscosity and a high density mean a higher turbulence and thus a higher heat transfer coefficient, they also mean more energy loss due to a higher pressure drop inside the tubes. The choice therefore depends on the trade-off between capital expenses and efficiency. (Podesser, 2003)

Ratio Critical Pressure to Condenser Pressure

A measure for telling whether a choice of a refrigerant will result in a high system COP is the ratio of its critical pressure to its condenser pressure. The higher this ratio, the higher the system COP. (Podesser, 2003)

Energy costs - Four refrigerants compared

The refrigerant properties determining the system efficiency are shown graphically for four different refrigerants in Figure 53. The heat pump system of Table 16 is used for the calculations.

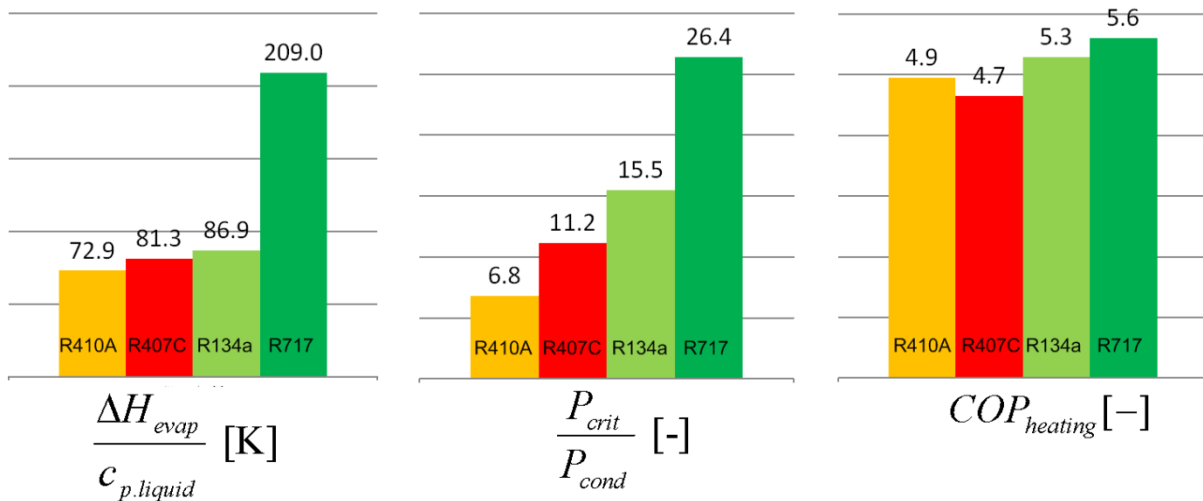


Figure 53. Refrigerant properties that determine system efficiency for four different refrigerants using the heat pump system of Table 16.

For all properties displayed in Figure 53 it is true that a higher value has a positive effect on the system efficiency (COP). It can be seen from Figure 53 that the ratio of latent heat of vaporization and critical pressure to condenser pressure give the right trend but do not tell the whole story when system efficiency is considered. Using R407C should result in a higher not a lower COP than using

R410A. Yet, this is not what COP calculations show (rightmost bar chart). The ratios do predict correctly that using ammonia should give the highest COP.

Conclusion

When comparing the results shown in Figure 52 and Figure 53 the following things can be noted:

- R134a has the least favourable volumetric heating capacity and condensation pressure and will therefore result in a less compact system and heavier system
- R717 has the best volumetric heating capacity and the lowest condensation pressure and will therefore give the most compact system
- R407C and R410A are somewhere in between when it comes to system compactness and weight
- R134a has the second best system efficiency
- R717 has the highest system efficiency
- R407C and R410A have the lowest system efficiency values, with R407C being a little worse than R410A. Their ideal COP's for heating mode are 4.65 and 4.95 respectively

From these results it is evident that R717 is a very good refrigerant. It results in a low system size and weight and a superior efficiency. Another advantage is that it does not contribute to global warming. This cannot be said for the three HFC refrigerants considered, which have Global Warming Potentials (GWPs) on a 100-year time horizon ranging from 1175 to 1725 kg CO₂-eq. The ozone depletion potential of all refrigerants in considered in this section is zero (IPCC, 2007). A downside to R717 is its toxicity. For this reason it has to comply with special regulations when it is used inside buildings. In the Netherlands the PGS 13 regulations give instructions on the indoor use of ammonia.

Since the results for cooling will show the same trend, it becomes clear why R134a is a commonly used refrigerant for household refrigerators: it has a high efficiency and a low toxicity. Since the required cooling capacity is relatively low for household use, the increased system weight is not too much of a problem. It furthermore has properties closely resembling those of R12. For this reason it became the refrigerant of choice when R12 was banned because of its ozone depleting properties.

Nowadays the refrigerants R407C and R410A are often used in air-source heat pump systems. The reasons for this are their high volumetric heating capacities and low normal boiling points. This last property important when outdoor temperatures drop. If the outdoor temperature is, for example, -20 °C and the temperature difference across the evaporator is 10 K, R134a cannot be used since its evaporating pressure will then drop below atmospheric pressure. R717 can be used, but since it is toxic it has to comply with special regulations, which limit its application in these systems. R410A and R407C suffer from none of these problems: their evaporating pressure is still higher than atmospheric pressure at -30 °C evaporating temperature and their toxicity is low.



Appendix I

Fits used for the heating power of the heat pumps in the heating system model

For the air source heat pump (Robur GAHP-AR):

- **Return temperature 20 °C, supply temperature 30 °C**

$$\dot{Q}_{\text{heating}} = -1.7265 \cdot 10^{-4} T_{\text{outdoor},in}^3 - 4.9508 \cdot 10^{-3} T_{\text{outdoor},in}^2 + 0.38859 T_{\text{outdoor},in} + 35.358$$

- **Return temperature 35 °C, supply temperature 45 °C**

$$\dot{Q}_{\text{heating}} = -4.0953 \cdot 10^{-4} T_{\text{outdoor},in}^3 - 2.6246 \cdot 10^{-3} T_{\text{outdoor},in}^2 + 0.54342 T_{\text{outdoor},in} + 33.424$$

- **Return temperature 40 °C, supply temperature 50 °C**

$$\dot{Q}_{\text{heating}} = -3.8523 \cdot 10^{-4} T_{\text{outdoor},in}^3 - 2.5437 \cdot 10^{-5} T_{\text{outdoor},in}^2 + 0.48789 T_{\text{outdoor},in} + 31.640$$

- **Return temperature 45 °C, supply temperature 60 °C**

$$\dot{Q}_{\text{heating}} = -2.4380 \cdot 10^{-4} T_{\text{outdoor},in}^3 - 8.4959 \cdot 10^{-4} T_{\text{outdoor},in}^2 + 0.40342 T_{\text{outdoor},in} + 30.147$$

For the ground source heat pump (Robur GAHP-GS):

- **Return temperature 25 °C, supply temperature 35 °C:**

$$\dot{Q}_{\text{heating}} = 0.54 T_{\text{source},in} + 42.6 \quad -5 \leq T_{\text{source},in} < 0$$

$$\dot{Q}_{\text{heating}} = 42.6 \quad 0 \leq T_{\text{source},in} < 5$$

$$\dot{Q}_{\text{heating}} = 0.02 T_{\text{source},in} + 42.5 \quad -5 \leq T_{\text{source},in} < 0$$

$$\dot{Q}_{\text{heating}} = 42.7 \quad T_{\text{source},in} \geq 10$$

- **Return temperature 35 °C, supply temperature 45 °C:**

$$\dot{Q}_{\text{heating}} = -2.4444 \cdot 10^{-4} T_{\text{source},in}^3 - 3.0000 \cdot 10^{-3} T_{\text{source},in}^2 + 0.32968 T_{\text{source},in} + 39.152$$

- **Return temperature 45 °C, supply temperature 55 °C**

$$\dot{Q}_{\text{heating}} = -6.6667 \cdot 10^{-5} T_{\text{source},in}^3 - 2.4286 \cdot 10^{-3} T_{\text{source},in}^2 + 0.35952 T_{\text{source},in} + 35.971$$

- **Return temperature 55 °C, supply temperature 65 °C**

$$\dot{Q}_{\text{heating}} = 0.3880 T_{\text{source},in} + 31.2400$$



For the water source heat pump (Robur GAHP-WS):

- **Return temperature 25 °C, supply temperature 35 °C:**

$$\dot{Q}_{\text{heating}} = 0.05T_{\text{source,in}} + 43.4 \quad 6 \leq T_{\text{source,in}} < 10$$

$$\dot{Q}_{\text{heating}} = 43.9 \quad T_{\text{source,in}} \geq 10$$

- **Return temperature 35 °C, supply temperature 45 °C:**

$$\dot{Q}_{\text{heating}} = 2.9204 \cdot 10^{-4} T_{\text{source,in}}^3 - 2.3335 \cdot 10^{-2} T_{\text{source,in}}^2 + 0.61741 T_{\text{source,in}} + 38.196 \quad (1.1.2)$$

- **Return temperature 45 °C, supply temperature 55 °C:**

$$\dot{Q}_{\text{heating}} = -9.7071 \cdot 10^{-3} T_{\text{source,in}}^2 + 0.57368 T_{\text{source,in}} + 34.347$$

- **Return temperature 55 °C, supply temperature 65 °C:**

$$\dot{Q}_{\text{heating}} = 6.8447 \cdot 10^{-4} T_{\text{source,in}}^3 - 4.2436 \cdot 10^{-2} T_{\text{source,in}}^2 + 1.0535 T_{\text{source,in}} + 28.558$$

Please note that the fits used above are valid for one heat pump. When using multiple heat pumps in series, the value for the heating power obtained from these equations has to be multiplied by the number of heat pumps.



Appendix J

Details on the heat demand profile calculations

The lines from Figure 23 have the following form:

$$\dot{Q}_{\text{heating,day}} = a_d T_{\text{outdoor}} + b_d \quad (1.1.3)$$

with

$$a_d = \frac{-\dot{Q}_{\text{max,day}}}{T_{\text{max}} - T_{\text{min}}} \quad b_d = \frac{\dot{Q}_{\text{max,day}}}{1 - T_{\text{min}} / T_{\text{max}}} \quad (1.1.4)$$

$$\dot{Q}_{\text{heating,night}} = a_n T_{\text{outdoor}} + b_n \quad (1.1.5)$$

with

$$a_n = \frac{-\dot{Q}_{\text{max,night}}}{T_{\text{max}} - T_{\text{min}}} \quad b_n = \frac{\dot{Q}_{\text{max,night}}}{1 - T_{\text{min}} / T_{\text{max}}} \quad (1.1.6)$$

And the relationship between $\dot{Q}_{\text{max,day}}$ and $\dot{Q}_{\text{max,night}}$:

$$\dot{Q}_{\text{max,night}} = f_{nd} \dot{Q}_{\text{max,day}} \quad (1.1.7)$$

The total yearly heating demand was calculated using data from SenterNovem (currently called Agentschap NL) on the average heating demand of nursing homes. This data, together with the other simulation parameters are given in Table 17. One might wonder why the heating power required at night is assumed to be half the heating power required during the day. There are various reasons for this. Firstly, the night time temperature setting is normally about 15 °C, which is 6 °C lower than the day time setting of 21 °C. This will result in reduced transmission losses. Secondly, because the building is still at 21 °C when the night begins, it takes quite some time for the indoor air to drop below 15 °C and it is only then that the heating system will have to provide for additional heating during the night. In the morning, the building has cooled down, meaning that the heating system during day time will have to provide a lot of extra heating power during the morning to warm up the large thermal mass of the building.



Table 17. Parameters used for calculating the heat demand profile of a typical nursing home (Dones, 2007), (SenterNovem, 2007).

f_{nd} [-]	0.5
Living area [m ²]	3000
Average heat demand [m ³ gas/m ²]	22
LHV Dutch nat. gas [MJ/m ³]	34.9
T_{min} [°C]	-10.5
T_{max} [°C]	18.0
Day starts at hour 8	
Day ends at hour 22	

The total yearly heating demand calculated using these parameters is:

$$Q_{tot,yearly} = (3000 \text{ m}^2)(22 \text{ m}^3 \text{ gas/m}^2)(34.9 \text{ MJ/m}^3) = 639833 \text{ kWh}_{th}$$

For every hour of the year the average heat demand is then estimated using equations (1.1.3) to (1.1.7). Since the peak heating demand ($\dot{Q}_{max,day}$) is initially unknown, an initial value has to be estimated. Calculations for one whole year are then performed, using hourly climate data from a suitable reference year from the KNMI as discussed in section 5.3. It is thereby assumed that the heating demand during one hour is constant, which is a simplification. The total yearly heating demand follows from those calculations by summation over all hours in one year:

$$Q_{tot,yearly,calc} = \sum_{t_{day}} \dot{Q}_{heating,day} \Delta t + \sum_{t_{night}} \dot{Q}_{heating,night} \Delta t$$

By setting $Q_{tot,yearly,calc} = Q_{tot,yearly}$ and letting Excel perform the iteration work, a value for $\dot{Q}_{max,day}$ can be computed:

$$\dot{Q}_{max,day} = 339 \text{ kW}$$

Appendix K

Influence of building heating water return temperature on the efficiency of the gas-fired boiler

The fits corresponding to Figure 26 on page 42 are:

$$15.51 \leq T_{w,building,out} \leq 56.10$$

$$\eta_{boiler} = -2.5872 \cdot 10^{-3} \cdot T_{w,building,out} + 1.0971$$

$$56.10 < T_{w,building,out} \leq 82.24$$

$$\eta_{boiler} = -1.2585 \cdot 10^{-3} \cdot T_{w,building,out} + 1.0225$$

